POWER PLANT ENGINEERING QUESTIONS AND ANSWERS

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POWER PLANT ENGINEERING QUESTIONS and ANSWERS

Giving full detailed information on the how and why of operation of power plant machinery

Revised and Arranged by
The Editorial Staff of
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CONTENTS

Part I. Boilers

Strength 17

1. Bursting pressure of shell; 2, working pressure of shell; 3, return tubular; water tube; 4, safe pressure for flat surfaces; 5, bursting pressure for firebox; 6, safe pressure for bumped heads; 7, area of segment; 8, segment area problem; 9, strain on longitudinal vs. roundabout joints; 10, figuring stress on longitudinal seam.

Boiler Joints 21

11. Efficiency of a joint; 12, figuring efficiency of double riveted lap joint; 13, average efficiency of seams; 14, strength of rivets vs. plate in joint; 15, strength of butt strap vs. lap joint; 16, problem, efficiency of double-riveted butt-strap joint; 17, why inside strap thicker and wider; 18, dimensions of lap joints; 19, rows of rivets in girth seam; 20, drilled rivet holes; 21, length of rivets; 22, spacing of rivets; 23, test for precooled rivet.

Boiler Tubes 27

24. Material and strength for tubes and flues; 25, spacing of tubes; 26, efficiency of ligament between tubes; 27, thickness of tubes; 28, plain and submerged vertical tubes; 29, beading tube ends; 30, short tubes; 31, beaded tubes in Stirling boiler; 32, curved tubes in Stirling boiler; 33, flared tubes in Heine boiler.

Staying of Boilers 32

34. Maximum area for one stay; 35, bracing below tubes; 36, bracing bumped heads; 37, staying of vertical firebox; 38, stay bolts, where used; 39, size of staybolts; 40, number of staybolts; 41, staybolt problem; 42, holes in staybolts; 43, strain on braces; 44, Cahall boiler bracing; 45, Edge Moor boiler bracing.

Material for Boilers.....
46. Parts of steel; 47, openhearth vs. Bessemer steel; 48, steel for marine boilers.

Fusible Plugs 38
49. Alloy for filling plugs; 50, location of plugs for tubular boilers; 51, for Heine boilers; 52, for Stirling boilers; 53, for Babcock & Wilcox boilers; 54, projection of plug; 55, effect of plug melting.

Boiler Settings 40
56. Setting for return tubular boiler; 57, advantage of high setting; 58, supporting return tubular boiler; 59, advantages of hangers; 60, slope of boiler; 61, thickness of side walls; 62, air spaces in walls; 63, Smith setting; 64, equalizer hanger; 65, space in rear of boiler; 66, covering for top of boiler; 67, support of Heine boiler; 68, of Babcock & Wilcox boiler; 69, bridge wall of B. & W.

Boiler Furnaces 47
70. Morrison furnace; 71, Fox furnace; 72, Purves furnace; 73, Leeds and Brown furnaces; 74, Adamson ring; 75, mean diameter of Morrison furnace; 76, pressure allowable on Morrison furnace; 77, location of bridge wall; 78, vertical vs. horizontal baffles.

Grates 50
79. Grate surface per hp.; 80, grate surface needed; 81, air space in grate; 82, ratio grate to heating surface.

Proportions and Construction..... 51
83. Capacity of boiler; 84, heating surface; 85, thickness of shell; 86, dry sheet; 87, flanging of head; 88, manhole frame; 89, pressure and diameter; 90, circulation in horizontal tubular boiler; 91, shells in sections; 92, small center course of shell; 93, location of gage cock; 94, heating surface of Manning boiler; 95, thickness of Manning water leg plates; 96, circulation in Manning water leg; 97, Manning vs. vertical tubular; 98, entrance of feed pipe in vertical boilers; 99, use of blow off; 100, pipe for blowoff; 101, size blowoff tank.

Boiler Operation 60

102. Leaky tubes; 103, repair of bagged boiler; 104, bag and blister; 105, thickness of metal in bag; 106, tensile strength of metal in bag; 107, leak at center seam; 108, kinds of patches; 109, soft patch on tube sheet; 110, effect of soot; 111, cleaning of tubes; 112, inspection for oil; 113, removing oil; 114, retubing B. & W. boiler; 115, leaking hand holes; 116, repair of cracked header; 117, re-ending tubes; 118, working boilers in parallel; 119, gage glass vs. try cocks; 120, proper water level; 121, cleaning out oil on water surface; 122, light vs. heavy load; 123, blowing down under pressure; 124, blowing down to wash out; 125, test for leaking blowoff; 126, clearing blowoff of scale; 127, cleaning mud drum; 128, cooling down return tubular; 129, reducing grate area; 130, methods of hand firing; 131, evaporation with hand firing; 132, saving by stoker; 133, draft over fire; 134, natural, forced, induced draft; 135, draft in battery of boilers; 136, Hawley down-draft furnace; 137, firing internal furnace; 138, stuck safety valve; 139, lap seam leak; 140, low water; 141, safety-valve setting for battery; 142, unstayed flat surfaces.

Boiler Inspection 75

143. External examination; 144, internal examination; 145, hydrostatic test; 146, inspection when taking charge; 147, watch inspection; 148, inspecting new boiler; 149, precautions when inspecting and cleaning; 150, safety for man inside boiler; 151, inspecting and cleaning a battery of boilers; 152, inspection when cleaning; 153, finding points of leakage; 154, locating broken stay bolt; 155, strength of pitted plate; 156, causes of explosions; 157, monthly report of condition.

Boiler Fuels 79

158. Horsepower from 1 lb. coal; 159, coal vs. oil; 160, mixed coal and oil; 161, conditions

for complete combustion; 162, air per 1 lb. carbon; 163, problem on air for coal; 164, air per 1 lb. fuel; 165, effect of insufficient air; 166, loss from incomplete combustion; 167, B.t.u. to heat air; 168, natural gas per horsepower-hour; 169, installation of gas burners; 170, natural gas as steam fuel.

Feed Water..... 85

171. Scale prevention; 172, cause of foaming; 173, preventing foaming; 174, remedy for foaming; 175, reason for priming; 176, soda ash needed; 177, removing oil with soda ash; 178, sal-ammoniac for lime; 179, result of poor circulation; 180a, internal rusting; 180b, oil and graphite for scale.

Feed Water Piping..... 89

181. Feed pipe entrance for B. & W. boiler; 182, feeding through blowoff; 183, feeding in battery; 184, vibration of feed piping; 185, valves on feed line; 186, lost valve disc; 187, stand pipe for feeding; 188, principle of injector; 189, double tube injector; 190, automatic injector; 191, hot water feed by injector; 192, injector with open and closed heaters; 193, injector capacity; 194, cleaning injector tubes; 195, feeding with lower pressure steam.

Accessory Apparatus..... 94

196. Size of safety valve; 197, pressure for lever safety valve; 198, action of pop valve; 199, action of open heater; 200, open heater and exhaust heating; 201, flooding engine from heater; 202, temperature with open heater; 203, flooding of heater; 204, types of closed heaters; 205, temperature with closed heater; 206, advantages of closed heater; 207, leaks in closed heater; 208, pumping with closed heater; 209, tubing for heater; 210, steam for heating tank; 211, economizer; 212, water column connections; 213, valves for water column; 214, gage glass renewal; 215, placing gage cocks; 216, action of gages; 217, testing steam gage.

Steam and Heating 106
218. Flow of steam; 219, velocity of jet; 220, condensation of steam; 221, limit of superheat; 222, superheat and pressure; 223, superheat in engines; 224, arrangement for exhaust heating; 225, vacuum heating; 226, size of steam and return pipes; 227, size of radiator; 228, condensation in radiator; 229, radiation and boiler required; 230, use of vacuum system; 231, disappearing water in heating system; 232, economy of high vs. low pressure; 233, heating of factory.

Boiler Efficiency 115
234. Causes of low efficiency; 235, calculation of efficiency from test; 236, evaporation per pound of fuel; 237, equivalent evaporation from and at 212 deg.

PART II. ENGINES AND TURBINES

Types of Engines 119
238. Advantages of high-speed engine; 239, of 4-valve engine; 240, of Corliss engine; 241, economical type; 242, Corliss and automatic cutoff; 243, single and double-acting; 244, square engine; 245, condensing vs. non-condensing; 246, variable lead types.

Pistons and Rods 122
247. Safe piston speed; 248, point of fastest piston travel; 249, angularity of connecting rod; 250, piston rings; 251, snap rings; 252, centering built-up piston; 253, advantages of built-up piston; 254, size of piston rod; 255, fastening piston rod; 256, pressure on guides; 257, centering small piston; 258, centering piston; 259, replacing follower bolts; 260, size of piston rings; 261, rounding off corners; 262, checking up rings; 263, replacing snap rings; 264, repair of built-up piston; 265, adjusting crosshead shoes; 266, oval crosshead pin.

Connecting Rod 132
267. Measuring length needed; 268, position for setting up crosshead shoes; 269, keying brasses and clearance; 270, keying up strap-end rod; 271, maintaining clearance.

Cylinders 135
272. Thickness of walls; 273, counterbore; 274, area of ports; 275, wear on bottom; 276, uneven wear; 277, finding clearance.

Engine Valves 138
278. Working edges of slide valve; 279, tight piston valves; 280, indirect piston valve; 281, balanced valve; 282, balancing slide valve; 283, lead of eccentric; 284, lift of poppet valve; 285, gear of Brown engine; 286, valve for shaft governor; 287, riding cutoff; 288, Meyer valve; 289 Corliss valve closure; 290, 2-eccentric Corliss; 291, longest Corliss cutoff; 292, 2-eccentric Corliss cutoff; 293, apparent and real cutoff and expansion; 294, lap and lead angles; 295, reason for lap and lead; 296, excess exhaust lap; 297, negative exhaust lap; 298, increasing angle of advance; 299, lead for high-speed engine; 300, changing advance vs. throw; 301, lap on Rollins engine; 302, earliest cutoff for slide valve; 303, why poppet valves used.

Valve Setting 148
304. Shortening slide valve cutoff; 305, changing lead; 306, increasing travel; 307, setting of slide valve; 308, of piston valve; 309, Corliss valves; 310, of double-eccentric Corliss; 311, of triple-expansion Corliss; 312, of Fitchburg valves; 313, of Buckeye valve; 314, of Greene-Wheelock valves.

Bearings 160
315. Length of main bearing; 316, taking up quarter boxes; 317, amount of take up; 318, adjustment of quarter boxes running; 319, babbiting crankpin brasses.

Eccentrics 163
320. Reducing eccentric diameter; 321, getting eccentric dimensions; 322, babbiting eccentric strap.

Indicator Cards 165
323. Carnot cycle; 324, Rankine cycle; 325, throttling and automatic cards; 326, indication of defects of setting; 327, excess com-

pression and leaky valve; 328, leaking piston; 329, slanting admission line; 330, figuring horsepower; 331, improving cards; 332, late events; 333, crooked expansion line; 334, faulty card; 335, defects of setting of angle-compound; 336, reducing motion; 337, unequal rocker travel.

Governors 179

338. Action of flyball governor; 339, of throttling and of shaft governor; 340, increasing weight of balls; 341, pendulum governor; 342, Gardner and Waters governors; 343, inertia governor; 344, shaft vs. throttling economy; 345, testing shaft governor; 346, broken governor spring; 347, broken inertia governor weight; 348, regulation for driving generator; 349, adjusting McIntosh & Seymour governor; 350, diameter of governor drive pulley; 351, changing Corliss speed; 352, hunting; 353, speed variation; 354, changing speed setting; 355, to decrease speed.

Power and Steam Consumption..... 189

356. Figuring horsepower; 357, horsepower constant; 358, increase in m.e.p.; 359, brake and indicated horsepower; 360, ratio of expansion; 361, problem on same; 362, expansion in compound engine; 363, increasing of power by cutoff; 364, increase of power with speed; 365, effect of increasing cutoff; 366, finding steam consumption; 367, steam from indicator diagram; 368, accuracy of preceding method; 369, steam per kilowatt-hour; 370, steam per stroke; 371, brake horsepwower.

Flywheels 194

372. Safe rim speed; 373, weight required; 374, problem on weight; 375, strain on bolts of joint.

Compound Engines..... 197

376. Advantage of condensing; 377, saving by compounding; 378, compounding vs. con-

densing; 379, when to use compound; 380, size of triple-expansion cylinders; 381, problem on preceding; 382, changing low-pressure cutoff; 383, expansions in compound; 384, terminal pressure; 385, regulating compression; 386, running on low-pressure side alone; 387, reason for receiver; 388, finding receiver pressure; 389, effect of receiver pressure; 390, increasing low-pressure load; 391, condenser pressure.

Operation of Engines..... 204

392. Changing Corliss cutoff; 393, to tell cutoff of Corliss; 394, dashpot rebound; 395, inspection of engine; 396, valve changes to run condensing; 397, hot crank pin; 398, crank pin pound and vibrating cylinder; 399, knock in bearing; 400, balancing engine; 401, test of alinement; 402, test of shaft for square with center line; 403, crosshead alinement; 404, testing crosshead shoes; 405, complete alinement test; 406, back pressure for heating; 407, one end condensing; 408, heating from receiver; 409, losses in efficiency; 410, excess back pressure; 411, unequal clearance; 412, using superheated steam; 413, faulty lubrication; 414, cylinder oil needed; 415, amount of cylinder oil.

Steam Turbines..... 223

416. Action of Curtis turbine; 417, action of Westinghouse turbine; 418, starting up turbine; 419, stopping turbine; 420, critical speed; 421, mixed pressure; 422, double flow; 423, pitch of blades; 424, end thrust; 425, velocity of steam; 426, Curtis vertical bearing; 427, failure of step lubrication; 428, sticking steam valves; 429, starting impulse vs. reaction; 430, action of regenerator; 431, Westinghouse governor; 432, Allis-Chalmers governor; 433, governing low-pressure turbine; 434, De Laval governor; 435, overspeed governor; 436, turbine steam consumption; 437, turbine horsepower; 438, use of high vacuum.

Part III. Auxiliary Steam Equipment

Condensers 234

439. Increase of horsepower; 440, action of jet and surface; 441, siphon and barometric; 442, length of tail pipe; 443, maximum vacuum; 444, vacuum and altitude; 445, vacuum and barometric pressure; 446, effect of drop in vacuum; 447, leak in barometric condenser; 448, lost vacuum; 449, condensing water needed; 450, dry vacuum pump; 451, use of copper tubes; 452, tube packing; 453, leaking tubes; 454, test for leakage; 455, leakage from condensate; 456, starting condensing plant; 457, flooding engine from surface condenser; 458, from siphon condenser.

Piping 243

459. Size of steam pipe for engine; 460, 90-degree bend; 461, 45-degree offset; 462, size of steam and exhaust pipes; 463, size for given amount of steam; 464, water hammer; 465, slope of steam line; 466, efficient separator; 467, exhaust oil extractor; 468, action of trap; 469, trap failure; 470, changing trap to high pressure from low; 471, Bundy trap; 472, repair of trap; 473, Holly loop; 474, thickness of pipe; 475, corrosion of steel and iron.

Reciprocating Pumps 250

476. Pump valves; 477, valve spring action; 478, valve lost motion; 479, spool adjustment; 480, steam valve lap; 481, duplex valve setting; 482, short stroke; 483, valve gear wear, 484, air chamber; 485, length of water cylinder; 486, piston striking; 487, tight rod packing; 488, running duplex on one side; 489, piston leakage; 490, simplex vs. duplex; 491, horsepower to drive pump; 492, head vs. steam pressure; 493, pump capacity; 494, jump at start of stroke; 495, vacuum pump for boiler feed; 496, dry and wet vacuum pump; 497, cost of pumping vs. city water; 498, steam expansion in pump; 499, high and low duty; 500, percent slip.

Centrifugal Pumps..... 258

501. Horsepower and discharge; 502, problem on driving motor horsepower; 503, belt-drive vs. direct steam; 504, delivery head; 505, pressure and runner; 506, diffusion rings; 507, turbine vs. reciprocating; 508, motor drive vs. steam turbine; 509, effect of closing discharge; 510, test of discharge flow; 511, pumping with heater; 512, efficiency, centrifugal vs. reciprocating.

Part IV. Combustion Engines

Principles and Construction..... 265

513. Operation of 4-stroke and 2-stroke cycle; 514, horsepower by A. L. A. M. rule; 515, heat per explosion; 516, heat in exhaust; 517, pressure after explosion, compression pressure; 518, effect of clearance; 519, test of exhaust gases; 520, gas per horsepower-hour; 521, ignition devices; 522, carburetor, use for gasoline and kerosene; 523, suction gas engines operating in parallel; 524, disposing of exhaust; 525, overheating; 526, muffling exhaust; 527, Diesel ignition; 528, economy, steam plant vs. oil engine; 529, gasoline drive for electric light and refrigeration.

Part V. Electrical Equipment

Direct Current Generators..... 276

530. Classes of d.c. dynamos; 531, use of series field shunt; 532, horsepower to drive generator; 533, use of equalizer; 534, starting compound-wound dynamo in parallel; 535, shutting down a dynamo from parallel operation; 536, cause of burnout; 537, reversal of polarity; 538, failure to generate; 539, remagnetizing field poles; 540, capacity of dynamos in series; 541, sparking and chattering; 542, sparking and burned bars; 543, burned mica; 544, armature reaction; 545, truing commutator; 546, rewinding field spool; 547, pitch of armature winding.

Alternating Current Generators 285

548. Poles, frequency and speed; 549, kilowatts vs. k.v.a.; 550, testing out phases; 551, induction alternator; 552, generator used as synchronous motor; 553, revolving field generator in parallel and as motor.

Motors 290

554. Opposed field windings; 555, broken field circuit; 556, speed and field strength; 557, calculating speed of d.c. motor; 558, horsepower of motor; 559, connections for d.c. compound-wound motor; 560, emergency starting box; 561, speed and load, series-wound motor; 562, use of series and shunt-wound motors; 563, d.c. generator as motor; 564, connections for squirrel-cage, induction motor; 565, rewinding induction motor; 566, starting compensator; 567, end thrust of rotor; 568, burnout of starting box.

Transmission 300

569. Three-wire system; 570, line self induction; 571, frequency to use; 572, testing electrostatic capacity; 573, test of insulation resistance; 574, test of strength of insulation; 575. Finding power factor of circuit; 576, p.f. of 4-wire and 3-wire system; 577, current per wire; 578, size of line wire; 579, problem on line wire; 580, smallest wire for circuit; 581, maximum load on lighting circuit; 582, constant-current transformer; 583, starting constant-current transformer; 584, cutting out series-lamp circuit; 585, maximum e.m.f. on constant-current transformer; 586, transformers in parallel; 587, ground on transformer lines; 588, transformer oil; 589, ratio of transformation; 590, current in induction coil; 591, horsepower in circuit; 592, resistance and current for parallel circuits; 593, electric heating; 594, phantom circuit; 595, opening lightning arrester; 596, corrosion of water pipes by a.c. and d.c.; 597, difference between a.c. and d.c.

Instruments 316

598. Voltmeter for a.c.; 599, output of 3-phase generator by wattmeter; 600, wattmeter constant; 601, light bill from wattmeter; 602, connections of synchroscope; 603, trouble in synchronizing a.c. generators; 604, connections for synchronizing lamps; 605, pull on a solenoid.

PART VI. REFRIGERATION 323

606. Principles of refrigeration; 607, boiling point of ammonia; 608, storage of ammonia; 609, ice melting and ice making capacity; 610, heat equivalent of 1 T. refrigeration; 611, rating of ice machines; 612, distilled water process; 613, symptoms of overload; 614, clearance in compressor; 615, single and double-acting compressors; 616, excessive clearance; 617, cause of compressor bursting; 618, compressors in parallel vs. separate; 619, checking capacity of compressors; 620, cooling compressor stuffing box; 621, control of head pressure; 622, power to drive compressor; 623, ammonia disappears; 624, ammonia in tank coils; 625, "boil over" in absorption system; 626, cloudy, tainted and brittle ice; 627, expansion of ammonia; 628, removing ammonia from absorption system; 629, needle expansion vs. flat disc valve; 630, defrosting coils; 631, temperature of condenser cooling water; 632, double-pipe vs. atmospheric condenser; 633, thawing brine cooler; 634, heat absorbed by brine; 635, direct expansion vs. brine coils, size of liquid and gas lines; 636, cooling milk; 637, cooling ice storage house; 638, temperature for food storage.

PART VII. MISCELLANEOUS

Compressed Air 345

639. Capacity of compressor; 640, oil for compressor cylinder; 641, leaky valves by indicator card; 642, free air for compressor; 643, air flow from orifice, power to compress air; 644, pressure and temperature.

Hydraulics 348

645. Flow of water through pipe; 646, velocity of water for given flow; 647, velocity and discharge of water; 648, flow through V notch; 649, flow through rectangular weir.

Belting and Gearing..... 350

650. Size driving pulley and belt; 651, belt for given horsepower; 652, pulleys for countershaft; 653, belt joint.

PART I.

Boilers.

Strength

1. Q.—What is the rule for the bursting pressure of a boiler shell?

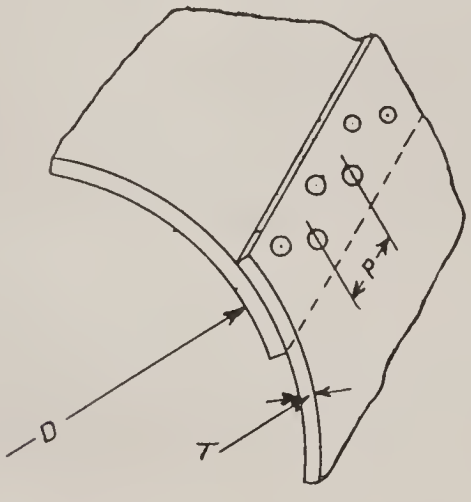
A.—The bursting pressure of a boiler may be found by the formula: $P = 2\ t\ S \div D$; where P = bursting pressure in pounds per square inch; t = thickness of shell, in inches; S = tensile strength, pounds per square inch; D = diameter of shell, inches.

2. Q.—Find the safe working pressure of a steam boiler, 50 in. in diameter ⅜-in. shell, double riveted longitudinal seams.

A.—Working pressure $Pw = 2\ S\ t\ e \div f\ D$

S (tensile strength) = 58,000
t (thickness of shell) = ⅜ in. or 0.375
e (joint efficiency) = 82%
f (factor of safety) = 5
D (diameter of shell) = 50 in.

$$\text{Working pressure} = \frac{2 \times 58{,}000 \times 0.375 \times 0.82}{5 \times 50} = 115 \text{ lb.}$$



A2. Safe working pressure in joint

3. Q.—How is the safe working pressure of a water-tube boiler determined?

A.—There is no standard method of determining the safe pressure of a water-tube boiler. Generally the

weak chamber in the boiler, so far as ordinary strength computations are concerned, is the steam drum. The strength for this is calculated the same as for the shell of a return tubular boiler. For other parts, the strength is generally much in excess of any requirements, and when a boiler gives way, it is usually by the failure of a screwed joint, a manhole or handhole bolt, or the fastening of a tube into a header.

4. Q.—How do you find the safe working pressure for flat unstayed surfaces?

A.—Few experiments have been made on the strength of flat unstayed heads, and the data so obtained has not been found satisfactory. One investigator has deduced the following rule, which was proved to be fairly accurate by actual tests: "To find the proper thickness for a flat unstayed head, multiply the area of the head by the pressure per square inch it is to bear safely, and multiply this by the desired factor of safety, say 8, then divide the product by ten times the tensile strength of the material used in the head." His rule for finding the bursting pressure when the dimensions of the head are given is: "Multiply the thickness of the head in inches by 10 times the tensile strength of the head material, and divide the product by the area of the head in square inches." The tensile strength may be taken at 44,800 lb. per square inch. One instance of a flat unstayed boiler head calculated by this rule was: Diameter of head 34.5 in., thickness 9/16, or 0.5625 in., tensile strength 44,800 lb. To find the bursting pressure, $0.5625 \times 44{,}800 \times 10 \div 935 = 270$ lb. per square inch bursting pressure. The head when tested to destruction burst at 280 lb. The safe working pressure of this head with a factor of safety of 8 would be $270 \div 8 = 33\frac{3}{4}$ lb. per square inch. (Keat's Mechanical Engineer's Pocket Book).

5 Q.—What is the pressure tending to rupture the outside sheet around a vertical boiler firebox?

A.—The pressure tending to rupture the external sheet around the firebox is the same as that tending to rupture this same sheet above the firebox, and is given in questions 1 and 2.

6. Q.—What is the safe working pressure for convex and concave bumped heads?

A.—In the A. S. M. E. Boiler Code, the safe working pressure of convex and concave heads is given by the formulas:

For convex heads, $P = (t - 1/8)\ S \div (R \times F)$

For concave heads, $P = 0.6$ the above

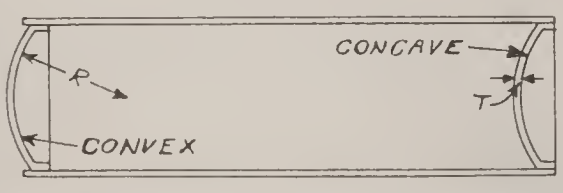
Where t = thickness of head in inches

S = tensile strength stamped on head

R = radius to which head is dished, inches

F = factor of safety, the lowest factor allowed in these rules being 5.5.

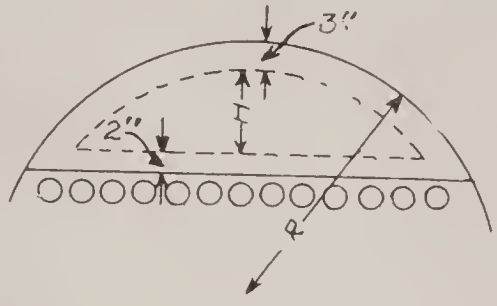
When a convex or concave head contains a manhole opening, the thickness should be increased by not less



A6. WORKING PRESSURE ON BUMPED HEADS

than 1/8 in., and in using the above formulas under these conditions for the effective thickness use t — 1/4 in.

7. Q.—How do you find the area of the section of a boiler head over the tubes?



A7. AREA OF STAYED SEGMENT

A.—The formula for finding the area of the segment of a circle is as given below:

$$A = \frac{4H^2}{3} \sqrt{\frac{2R}{H} - 0.608}$$

Where H is the distance from the chord of the circle to the outside, and R is the radius of the circle. In the case of a boiler head, the area to be stayed above the

tubes is found by the same formula, only that the value of H is taken as the height of the segment less 5 in., and the radius is taken as the radius of the shell less 3 in.

8. Q.—In a 72-in. boiler, the height of the segment to be stayed is 16 in. What is the area of this segment?

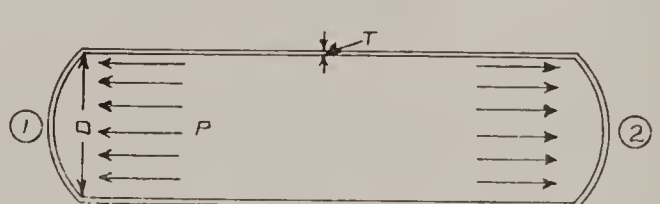
A.—In determining the area to be stayed, it is customary to assume that the shell will stay that portion of the area within 3 in. of the shell, and the top row of tubes will stay that portion of the area within 2 in. of the tubes, as shown dotted in the sketch, A7. The area to be stayed, then, is between the dotted lines, and is 16 in. in height on the center line of the boiler. Using the formula of Answer 7,

$$\frac{4 \times 16^2}{3} \times \sqrt{\frac{72}{16} - 0.608} = (341.3 \times 1.97) = 675.4$$

sq. in. area of segment to be stayed.

9. Q.—Why is the strain greater on the longitudinal joint than on the circumferential joint of a boiler shell?

A.—Assuming D = the diameter of boiler in inches, P = the pressure in pounds per square inch, t = the thickness of plate in inches, S = the stress in the material in pounds per square inch; then for a longitudinal section, the total pressure on a unit length, say one inch, equals $P \times D \times 1$, Fig. A10. This pressure is resisted by two sections of plate, t inches in thickness by one inch long. Then the pull in the material,



A9. STRESS ON CIRCUMFERENTIAL SECTION

$2t \times S = P \times D \times 1$, the total pressure exerted on it. Dividing both sides of the equation by 2t gives:

$$S = P \times D \div 2t \qquad (1)$$

Considering a circumferential section, the total pressure on the end, pressure per square inch times area, is $P \times 3.14 \times D \times D \div 4$. This is resisted by a section of plate, t inches in thickness by the circumference,

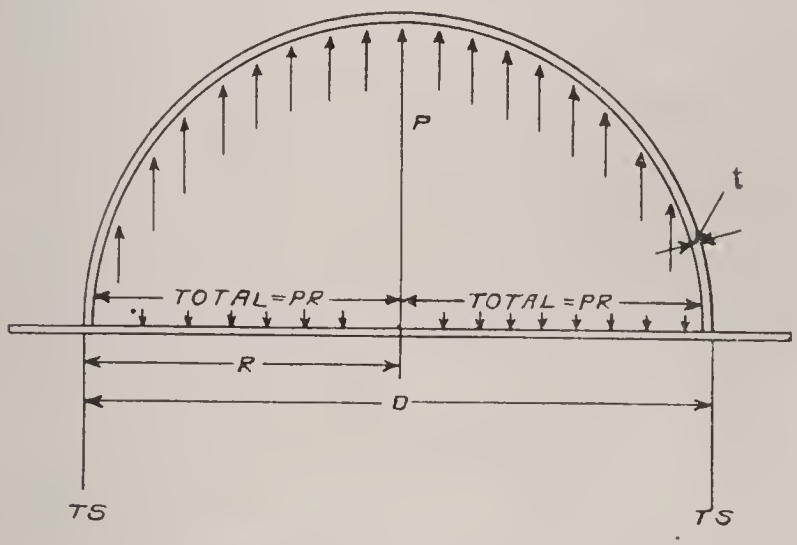
3.14 × D inches long = 3.14 × D × t sq. in. Then the resistance in the material, S × 3.14 × t × D = P × 3.14 × D × D ÷ 4. Dividing both sides of the equation by 3.14 × t × D gives:

$$S = p \times D \div 4t. \qquad (2)$$

Since the value of S obtained for a girth section (1) is one-half that for a longitudinal section (2), the strain on a longitudinal joint is twice that on a circumferential joint.

10. Q.—In figuring stress on the longitudinal seam of a boiler, why is the radius, or half the diameter taken?

A.—Suppose that instead of a complete cylinder you had a half cylinder 1 in. in length with a heavy



A10. PRESSURE TO RUPTURE SHELL

flat plate riveted across the diameter, as in Fig. A10. Evidently the pressure tending to force this plate off is equal to the pressure per square inch times the diameter of the cylinder, and this pressure is resisted by the pull on the two sides of the cylinder.

We can, therefore, in finding the stress, multiply the diameter by the pressure and divide this by twice the thickness of the plate, or we can multiply the pressure by half the diameter and divide by the thickness of the plate, to find the value of S.

Boiler Joints

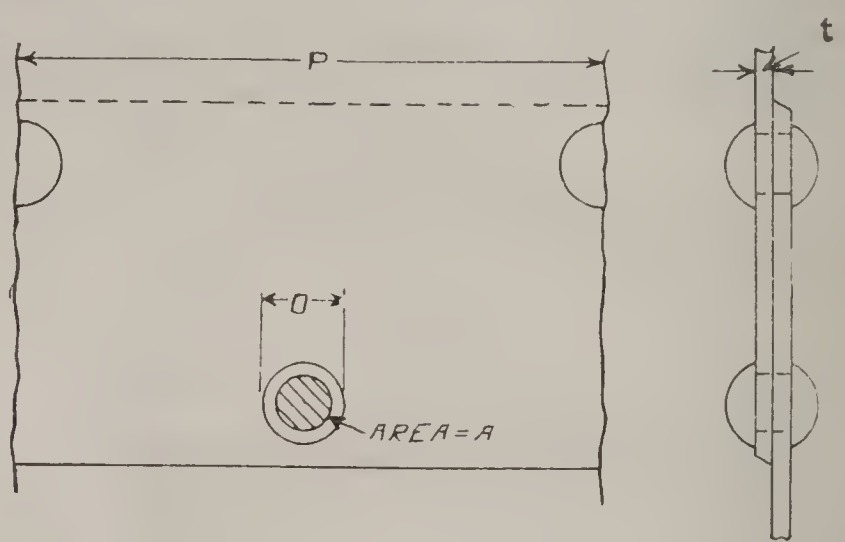
11. Q.—What is the "efficiency" of a boiler joint, and how is it found?

A.—The efficiency of a boiler joint is the strength of a unit length of the joint divided by the strength of

the same unit length of solid plate. In some riveted joints, failure is possible in a variety of ways, and to determine the efficiency, the strength of the weakest part of the joint must be divided by the strength of the solid plate.

12. Q.—How would you figure the efficiency of a double-riveted lap joint of the following dimensions:

Thickness of sheet, 3/8 in.; diameter of rivet holes, 15/16 in.; pitch of rivets, 3¼ in.; tensile strength of plates 55,000 lb. per sq. in.; shearing strength of steel rivets, 42,000 lb. per sq. in.



A12. EFFICIENCY OF DOUBLE-RIVETED JOINT

A.—In order to solve this problem, it will be necessary to calculate the strength of the solid plate, the strength of the plate between the rivet holes, the shearing strength of two rivets in double shear, and the crushing strength of the plate in front of two rivets.

The strength of the solid plate $= p \times t \times TS$.

The strength of the plate between the rivet holes $= (p - d) \times t \times TS$.

The shearing strength of rivets $= n \times s \times a$.

The crushing strength of the plate in front of rivets $= n \times d \times t \times c$.

The efficiency of the joint is then found by dividing the smallest of the results obtained for the last three formulas above, by the strength of the solid plate.

In the above formulas:

p = pitch of rivets in inches on the row having the greatest pitch;

t = thickness of the plate, in inches;

TS. = tensile strength, in pounds per square inch;

d = diameter of the rivet after driving, or the diameter of the rivet hole;

n = number of rivets in single shear per unit length of joint;

s = shearing strength of rivets, which for iron we can take as 38,000 lb. per square inch, and for steel, 42,000 lb. per square inch;

a = the cross-sectional area of the rivet after driving;

c = the crushing strength of mild steel, which we can take as 95,000 lb. per square inch.

Substituting the values given in your problem, we have, $3.25 \times 0.375 \times 55{,}000 = 66{,}990$ lb. as the strength of the solid plate.

The strength of the plate between rivet holes is equal to $(3.25 - 0.94)\ 0.375 \times 55{,}000 = 47{,}300$ lb.

For a double riveted joint there are two rivets per unit length and the shearing strength of two rivets is, $2 \times 42{,}000 \times 42{,}000 \times 0.94^2 \times 0.7854 = 57{,}120$ lb.

The crushing strength of the plate in front of two rivets $= 2 \times 0.375 \times 95{,}000 \times 0.94 = 66{,}500$ lb.

From the last three results we find that the strength of the plate between the rivet holes is smaller than either of the other two, so that we divide the value of this, which we have found to be 47,300, by the value of the strength of the solid sheet, or 66,990 lb., and obtain as a result 0.706, which multiplied by 100 is equal to 70.6 per cent, the efficiency of the joint.

13. Q.—If two seams of a boiler figure different efficiencies, can the average of the two be used in figuring the safe working pressure of the boiler?

A.—In figuring the efficiency of the seams of a boiler, whether these seams are single or double riveted, it will be necessary, when calculating the safe working pressure of the boiler, to take the lower efficiency, and not the average of the two. You can realize this to be true when you consider that "no chain is stronger than its weakest link." It is, of course, the same with a steam boiler; it is no stronger than its weakest seam.

14. Q.—Why does the strength of the rivet section

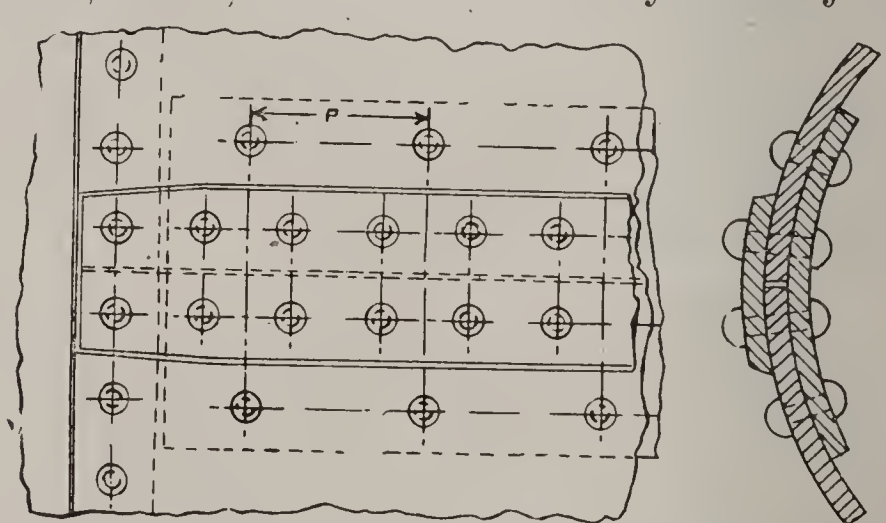
of a joint always figure greater than that of the plate section?

A.—The fact that the rivet section of a joint figures stronger than the plate between the rivets in the outer row, is due to the proportion and design of the joint. In a properly designed joint, the weakest part is the plate between the rivets in the outer row. But a joint may be so designed that the plate strength is greater than that of the rivets, if small rivets and large pitch are used.

15. Q.—What advantage has a double butt-strap joint over a lap joint?

A.—The butt and double strap joint is stronger than the lap joint, and it permits the making of a round boiler shell, which is of more importance than the extra 10 or 15 per cent of strength of the butt joint, over the ordinary double riveted lap seam.

16. Q.—In a double zigzag riveted butt joint, with two cover plates, the pitch of the rivets is 4⅛ in. The rivet holes are 15/16 in. diameter, and the plate is ¾ in. thick. If the ultimate tensile strength of the plate is 58,000 lb., and the shearing strength of the rivets is 40,000 lb., what is the efficiency of the joint?



A16. DOUBLE BUTT-STRAP JOINT

A.—Using the same symbols as in Answer 12;
TS. = 58,000 lb. = tensile strength of plate;
t = ¾ in. = 0.75 = thickness of plate;
p = 4⅛ in. = 4.125 = pitch of rivets;
d = 15/16 in. = 0.9325 = dia. of rivet after driving;

a = 0.6830 = cross-sectional area of rivet in square inches;

s = 40,000 lb. = shearing strength of rivets.

$p \times t \times TS = 179,452$ = strength of solid plate of unit length.

$(p - d)\ t \times TS. = 138,852$ = strength of plate between rivet holes.

$138,852 \div 179,452 = 0.774$ = efficiency of joint for plate strength.

For shearing of rivets, $n \times s \times a = 4 \times 40,000 \times 0.6830 = 109,280$ lb.

Efficiency $= 109,280 \div 179,\ 452 = 0.61$.

For crushing of plate, $n \times d \times t \times c = 2 \times 0.9325 \times 0.75 \times 95,000 = 132,867$ lb.

Efficiency $= 132,867 \div 179,452 = 0.744$.

The smallest efficiency is that of rivets in shear, = 61 per cent.

17. Q.—Why is the inside strap of a butt-strap joint thicker and wider than the outside one?

A.—The inside butt strap is thicker and wider than the outside because it is intended to take the tensile strain to a considerable extent, and also to stiffen the joint against bending; while the outside strap is intended more to hold down the edges of the sheet.

With the heavier and thicker strap inside, it is easier to make the edges of the sheet conform to the curvature of the strap than it would be if the straps were reversed or were alike. It would be possible, of course, to make the straps alike, but experience has shown that the present method of construction is better mechanically, and allows of better calking.

18. Q.—What are good dimensions for single and double riveted lap joints and sizes of rivets for 5/16, 3/8 and 7/16-in. plates?

A.—From Parson's "Steam Boilers" we get the following figures on single- and double-riveted lap joints:

Thickness of plate, in.	Diameter of rivet, in.	Pitch, in.	
		single riveted	double riveted
5/16	5/8	1 5/8	2 1/8
3/8	3/4	1 7/8	2 1/2
7/16	13/16	2	2 3/4

19. Q.—How many rows of rivets are used in the girth seam of a boiler?

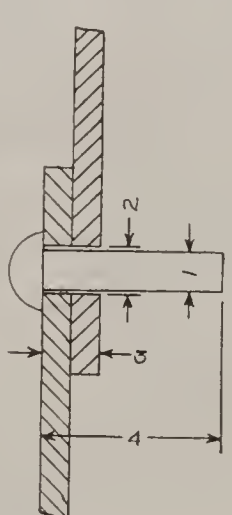
A.—Only one row is used because, properly designed, a single riveted seam has 56 per cent of the strength of the solid sheet and this is in excess of any stress that can be brought on this seam, as the total pressure tending to pull the shell apart at the girth seam is only one-half that on the longitudinal seam. (See answer 9.)

20. Q.—Why should rivet holes be drilled rather than punched?

A.—Drilled holes for rivets are better than punched because the plate is injured more or less in the process of punching, which sets up undue strains. Punched holes are never used in boiler work unless the hole is punched under size and then reamed to proper size, in order to remove the weakened material, and any cracks started at the edge of the holes by punching.

21. Q.—How long should a rivet be before it is driven to form the proper size head?

A.—The length of rivet before it is driven depends, of course, upon the size of the rivet, the size of the hole



A21. LENGTH OF RIVET BEFORE DRIVING

and thickness of the joint. Complete tables have been made up for the use of boiler manufacturers, giving the length of rivets for various sizes of joints, and the variation of the length may be noted from the following data taken from one of these tables, which will give you an idea as to the allowance for filling the hole and making the head:

(1) Diam. Rivet, In.	(2) Diam. Hole, In.	(3) Thickness of Joint, In.	(4) Length of Rivet Before Driv., In.
5/8	21/32	1/2	1 5/8
5/8	21/32	3/4	1 7/8
5/8	11/16	1/2	1 5/8
5/8	11/16	3/4	2
5/8	21/32	9/16	1 5/8
5/8	21/32	27/32	2
5/8	11/16	9/16	1 3/4
5/8	11/16	27/32	2
1 5/8	1 11/16	2 9/16	5 1/2
1 5/8	1 11/16	3 5/8	6 5/8

Boiler rivets when driven must completely fill the hole in which they are put, and form a head which will have the same strength as the tensile strength of the body of the rivet.

22. Q.—Why are small, closely spaced rivets used in a boiler joint instead of widely spaced large ones?

A.—Two things must be accomplished in the boiler joint: First, to keep the joint tight so that it can be calked to hold pressure; second, to hold the parts together. While the strength against tearing apart or shearing might be accomplished with the large rivets widely spaced, the edges of the sheet would buckle so that the pressure would force the sheets apart.

23. Q.—How can you tell whether a rivet has cooled before it was set up so as to bind the plates together?

A.—If a rivet cools off before being set up, a space will be left between the rivet and plate, or between the two plates. This can usually be detected by tapping the rivet lightly with a hammer.

Boiler Tubes

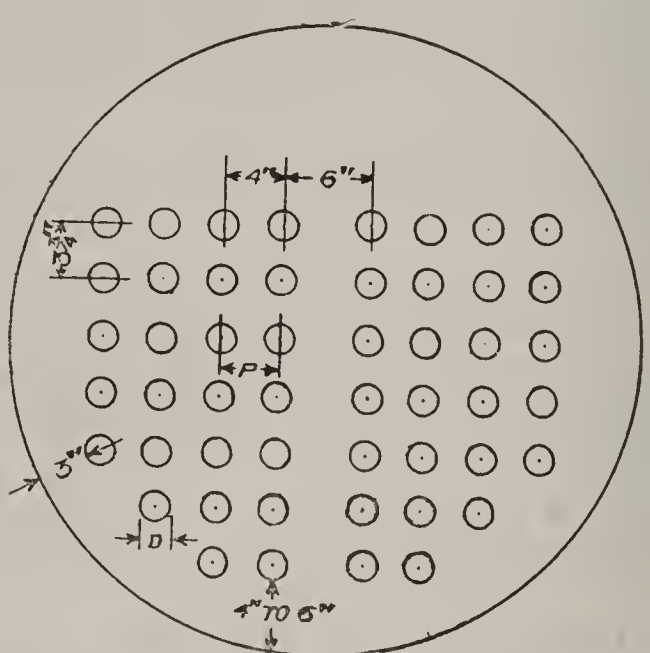
24. Q.—Of what are boiler tubes and flues made? What is the tensile strength? How many grades are used?

A.—Boiler flues are usually made either of steel or wrought iron. For the steel, tensile strength is usually taken 55,000 lb. per square inch, and for wrought iron, 45,000 lb. Only one grade is permitted for boiler flues, open hearth steel or knobbled hammered charcoal

iron. Seamless tubes must be of open hearth steel but may be either "flange" or "firebox" grade.

25. Q.—In placing 3-in. tubes in the head of a steam boiler, what size bridges should be left to allow for proper amount of heating surface and circulation?

A.—In a horizontal tubular boiler using 3-in. tubes, the tubes are spaced 4 in. between centers horizontally, and 3¾ in. between centers vertically with the two center rows 6 in. between centers. No tube should be nearer than 3 in. to the shell, and the bottom row should be from 4 in. to 6 in. from the bottom of the boiler. Ordinarily from 1 to 1½ in. between the outsides of tubes is considered desirable.



A25. SPACING OF TUBES

26. Q.—How do you figure the efficiency of the ligament between the tubes in the head of a return tubular boiler?

A.—When the pitch of the tube holes on every row is equal, Fig. A25, the formula is:

$$\frac{p - d}{p} = \text{efficiency of ligament}$$

where p = pitch of tube holes, inches
d = diameter of tube holes, inches

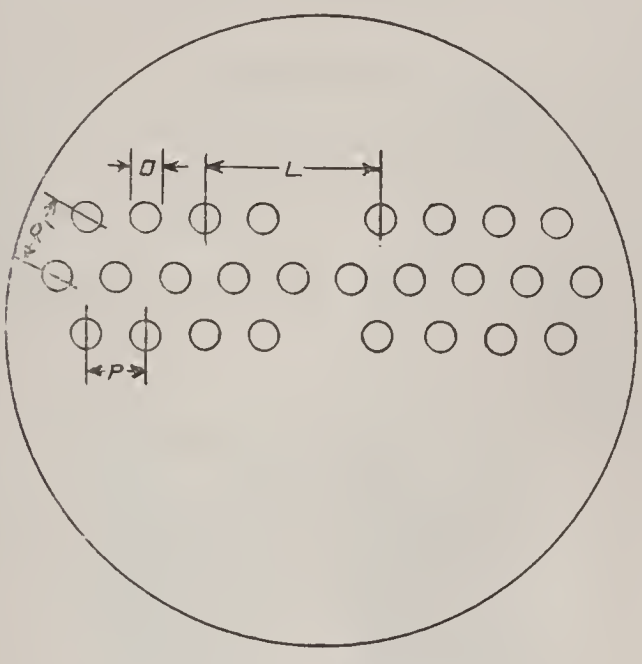
As an example: Assume the pitch of tube holes in the drum to be 5¼ in., the diameter of tubes 3¼ in., and the diameter of tube holes, 3 9/32 in. We then have,

$$\frac{p - d}{p} = \frac{5.25 - 3.281}{5.25} = 0.375, \text{ efficiency of ligament.}$$

When the pitch of tube holes on any one row is unequal, Fig. A26, the formula is:

$$\frac{l - nd}{l} = \text{efficiency of ligament}$$

where l = unit length of ligament, inches
n = number of tube holes in length, l
d = diameter of tube holes, inches.



A26. TUBES SPACED DIAGONALLY AND UNEQUAL PITCH

As an example let us assume having the unit length of ligament, in inches, equal to 12, the number of tube holes in this length, 2, and the diameter of the tube holes, in inches, equal to 3.281. Substituting these values in the formula just given, we have,

$$\frac{12 - 2 \times 3.281}{12} = 0.453 \text{ as the efficiency of the ligament.}$$

When a shell or drum is drilled for tube holes in a line diagonal with the axis of the shell or drum, Fig. A26, the efficiency of the ligament between the tube holes is determined by the following methods and the lowest value used:

(a) $$\frac{0.95\ (p_1 - d)}{p_1} = \text{efficiency of ligament}$$

$$(b)\ \frac{p - d}{p} = \text{efficiency of ligament}$$

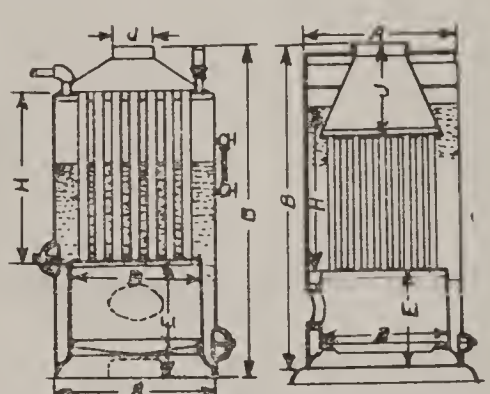
In these two formulas, p_1 is the diagonal pitch of tube holes, in inches; d is the diameter of tube holes, in inches; p is the longitudinal pitch of tube holes or distance between centers of tubes in a longitudinal row, in inches.

The constant 0.95 in formula (a) applies provided $p_1 \div d$ is 1.5 or over.

27. Q.—What should be the thickness of 3½-in. boiler tubes?

A.—The standard thickness of a 3½-in. boiler tube is 0.12 in., or a little less than ⅛ in. Extra strong tubes have a thickness of 0.32 in. in this size, but these tubes are not often used in ordinary practice.

28. Q.—What is the difference between plain and submerged tubes?



A28. PLAIN AND SUBMERGED VERTICAL TUBES

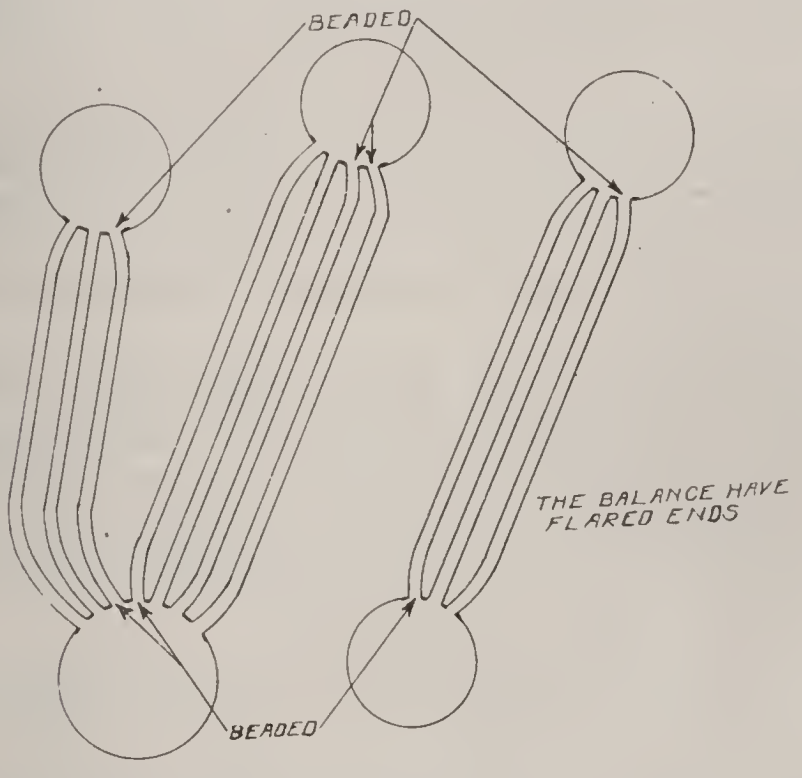
A.—In vertical fire-tube boilers we have two types: the ordinary type, which has the upper ends of the tubes in the steam space, having the hot gases on the inside and steam on the outside, which would absorb the heat, making superheated steam.

The submerged type has all the tubes totally submerged by having the upper tube sheet below the water line, and connecting it with a smoke flue to the upper head. In this way the entire length of the tubes is surrounded by water.

29. Q.—Is it better to bead over tube ends or not?

A.—Although it is not always done, it is better, if possible, to bead the tubes in any boiler. This is espe-

cially true in the case of a Stirling water-tube boiler, Fig. A29. If the tubes entering the lower part of the upper drums are not beaded, there will be a tendency for the water to collect in the drums after the boiler has been drained preparatory to cleaning, and it will be almost impossible to remove that water in any other way than to dip it out by hand. Those tubes entering the upper section of the mud drum should also be beaded, because if this is not done there will be a



A29. BEADING OF TUBES IN A STIRLING BOILER

tendency to form an air pocket as the boiler is being filled with water, and trap the air between the upper part of the drum and the surface of the contained water.

30. Q.—If tubes ordered for a return tubular boiler came ¼ in. short, what could be done with them?

A.—If boiler tubes were ordered with the intention of beading them, and they came too short by ¼ in., they could still be used by rolling them and flaring the end out with the peen of a hammer, as many boilers operate regularly with flared tubes.

In water-tube boilers the tubes are seldom, if ever, beaded, while in railroad practice, flaring is becoming more and more common. Tests have been made with boilers in which the tubes have been set by the several methods, and it was found that a flared tube withstood

a slightly higher pressure than a beaded tube when tested under hydraulic pressure to destruction. A number of tests showed very close to this general result.

31. Q.—Are any tubes beaded in the Stirling boiler? Where and why?

A.—Every other tube in a Stirling boiler is beaded over. In the upper drum, this beading does away with the settling of water which is liable, with the tubes flared, after the boiler has been blown down ready for cleaning. This arrangement drains the drums very nicely.

In the bottom or mud drum, it does away with the air lock, which is apt to form between the tubes with all the ends flared and not beaded over. See Fig. A29.

32. Q.—Why are the tubes of Stirling water-tube boilers curved?

A.—The tubes in the Stirling boiler are curved at the ends so that they may enter the drums at right angles to the drums. With the tubes set as they are on an incline, if the ends were straight, they would come into the drum at an angle to the shell, and it would be impossible to make a tight joint.

33. Q.—Why are the tubes of a Heine boiler flared after having been rolled?

A.—The tubes of a Heine or other water-tube boiler are usually flared to provide additional strength to the joint.

Staying of Boilers

34. Q.—A boiler has heads made of ¾-in. plate; the steam pressure is 100 lb. Compute the maximum area that can be safely supported by one stay.

A.—Rules of the A. S. M. E. Boiler Code give as maximum pressure to be carried on a stayed surface, with uniform stays symmetrically spaced:

$$P = C \times t^2 \div p^2$$

where P = max. working press. lb. per sq. in.
t = thickness of plate in *sixteenths* of an inch,
p = pitch of stays, inches,
C = a constant; 112 for stays screwed through and riveted over, plates not over 7/16 in.; 120 for screwed and riveted stays, plates over 7/16 in. thick; 135 for stays screwed through

and a single nut outside the plate; 175 for stays with inside and outside nuts and outside washers whose diameter is not less than 0.4 p and thickness not less than t.

Solving the above for pitch, we have: $p = t \sqrt{C \div P}$ and, assuming the third condition for C gives,

$$p = 12 \sqrt{135 \div 100} = 13.9 \text{ in.}$$

The area supported would be $p^2 = 194$ sq. in.

35. Q.—Why are braces from head to shell never used on horizontal tubular boilers below the tubes?

A.—In horizontal tubular boilers, it is not necessary to have braces from the head to the shell below the tubes, for the reason that the tubes themselves perform the function of braces.

36. Q.—Why are not braces used in a bumped head?

A.—A bumped head does not need staying, because the tendency of pressure is to force the head of the containing vessel into spherical form, and as the bumped head has this to a considerable extent, the pressure inside does not tend to distort it.

37. Q.—In a vertical boiler, which sheet of the fire box has to be stayed, the inside or outside?

A.—In a vertical or upright boiler the fire box sheet has to be stayed in order to keep it from collapsing. The fire box sides are stayed to the shell of the boiler by bolts screwed through both sheets and riveted over. The tubes act as stays for the crown sheet and upper head.

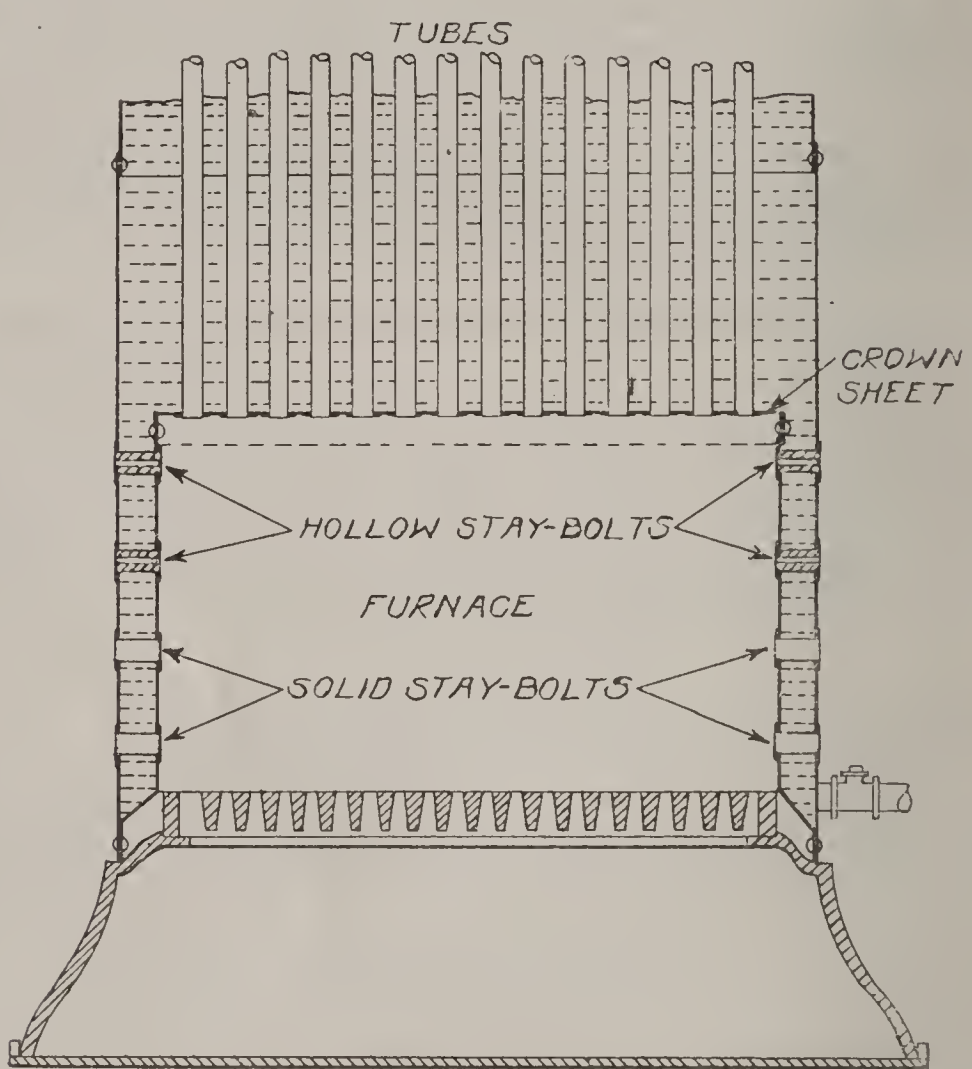
38. Q.—What are stay bolts, and what part of a steam boiler are they used in?

A.—Stay bolts are, or should be, made of the best quality of wrought iron, their diameter being ¾ in., ⅞ in. or 1¼ in., depending upon the size of the boiler, and the pressure it is to carry. They have a screw thread cut their entire length, and are screwed through both the outside and inside plates of fire box boilers at intervals of 4 to 4½ in. from center to center, thus securely binding the plates together. The ends are allowed to project through the plates far enough to permit of their being riveted down onto the plates. In some cases they are made hollow, so that a broken bolt

will be shown by leakage. To avoid threading the entire length, the ends are sometimes upset.

39. Q.—How is the proper size of stay-bolt determined?

A.—To determine the proper size of a stay-bolt, first multiply the pressure in pounds per square inch by the square of the pitch of the stay-bolts. This gives the total pressure which each stay-bolt has to carry.



A38. STAYING OF VERTICAL FIRE BOX

Divide this total pressure by the allowable stress per square inch on the stay-bolt, and it gives the area of the bolt at the bottom of the thread. From this area find the size of bolt permissible. The allowable stress per square inch for stay-bolts 1¼ in. diameter and under, according to the Massachusetts Rules, is 6500 lb.; over 1¼ in. diameter, 7000 lb.

40. Q.—How would you find the diameter and the number of head stays required for a boiler of any size?

A.—In order to find the size and number of stay-rods required in a boiler, it is first necessary to find the area of the surface to be stayed, as in answer 7.

Having found the area of segment, multiply by the pressure in pounds per square inch, in order to get the total pressure on this area.

The pitch of stays depends upon the thickness of the plate to be supported, and the maximum pressure to be carried. Assuming the stays are pitched 14 in. apart horizontally and 13 in. vertically, each stay would be required to support an area $14 \times 13 = 182$ sq. in., and the number of stays would be the area to be stayed divided by 182 sq. in.

Assuming the pressure to be 100 lb. per square inch, the total stress on each stay equals $182 \times 100 = 18{,}200$ lb. Assume the stay rods to be of soft steel having a tensile strength of 50,000 lb. per square inch, and using a factor of safety of 8, the sectional area required for each stay will be as follows: $18{,}200 \times 8 \div 50{,}000 = 2.9$ sq. in., and the diameter will be 1.9 in., or practically 2 in.

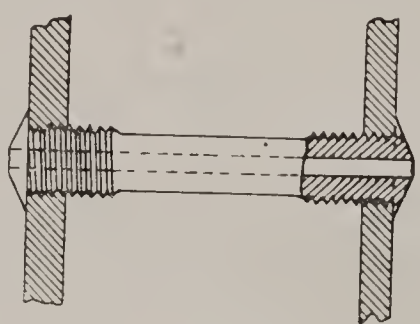
41. Q.—How would you find the size and number of stay-bolts needed to support an area of 20 by 32 in. of $\frac{3}{8}$-in. plate to carry 120 lb. per sq. in. working pressure?

A.—According to the Massachusetts Boiler Rules, the maximum pitch allowed for stay bolts in $\frac{3}{8}$-in. boiler plate, under pressure of 120 lb. is $5\frac{3}{4}$ in. For a 20-in. by 32-in. area, a pitch of $5\frac{1}{2}$ in. is more convenient than $5\frac{3}{4}$ in., and using six stay-bolts in a row, and four rows, makes 24 bolts. The pressure on this area equals $20 \times 32 \times 120$ lb. $= 76{,}800$ lb.

Assuming an allowable stress of 6500 lb. per square inch for stay bolts of mild steel, the net area of each stay should be equal to $20 \times 32 \times 120 \div 24 \times 6500 = 0.493$. With a V-thread of 12 threads per inch, the net area of a 15/16-in. stay equals 0.494. Therefore, 24 15/16-in. stays will support the 20 by 32-in. area under a pressure of 120 lb. An equivalent area of stayed surface would be supported by 20 stays of about 1 1/16 in. diameter and a pitch of $5\frac{3}{4}$ in.

42. Q.—Where should telltale holes be placed in stay bolts, and why are they recommended?

A.—The use of holes in stay bolts is to show if the stay bolts become cracked or broken. The holes are drilled straight through the center of the bolts lengthwise, and thus, if any break of a bolt occurs, the water will be forced out through this hole. It requires, of course, a somewhat larger stay bolt than when a solid bolt is used, thus cutting into the strength of the plate a little more, and also decreasing the flexibility of the

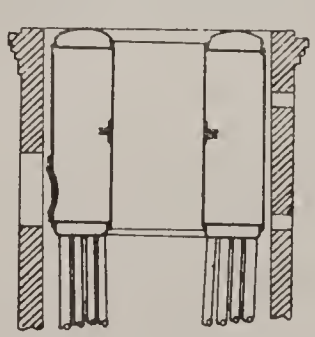


A42. SECTION OF HOLLOW STAY BOLT

stay bolt, but it is found desirable where the best results are looked for. The expense is somewhat greater than for solid bolts.

43. Q.—How could you tell what brace had the most strain on it in a horizontal tubular boiler?

A.—It would be absolutely impossible to tell what brace had the most strain on it. You could figure out which one was carrying the greatest load, if you knew the exact pressure at all points, and the amount of sur-



A44. SECTION OF CAHALL BOILER

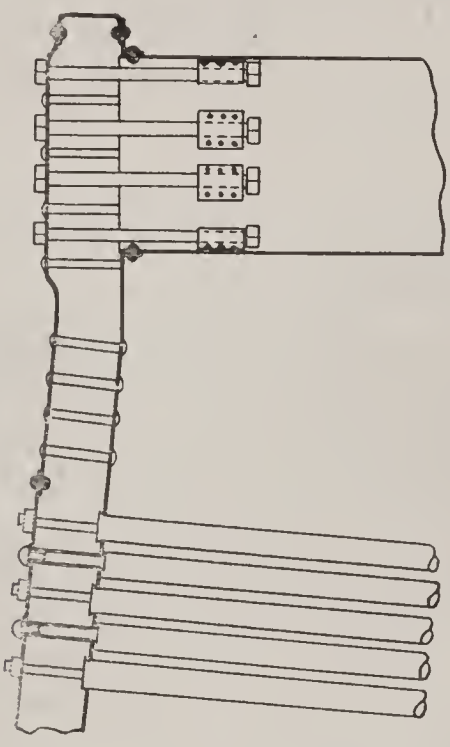
face which each brace supported, but in boiler design, all braces are supposed to be arranged to take equal amounts of load, and the way in which this load will actually be distributed in practice cannot be determined, because one brace may be a trifle longer than another and not take its load until after the short one has been stretched or the head has been flexed. By tapping with a hammer, the brace under greatest tension would tend

to give out the sharpest sound, but conditions might be changed when the boiler is under pressure.

44. Q.—How is the Cahall water-tube boiler braced?

A.—In a Cahall vertical water-tube boiler there are braces in the upper drum, extending from the bottom to the top. These are through braces, with "Y" connections riveted to the heads. No brace or stiffness is provided in the lower ring.

45. Q.—How is the Edge Moor boiler braced?



A45. BRACING OF EDGE MOOR BOILER

A.—The bulges of the headers opposite the drums of Edge Moor boilers are effectively stayed by eight 2½-in. steel bolts anchored through steel "U" plates riveted to the inside of the drum.

Material for Boilers

46. Q.—What parts of a boiler have to be of steel?

A.—The shells, drums and butt straps of boilers are made of open-hearth fire-box steel.

Heads, combustion chambers, furnaces, or any plates that require staying or flanging, are made of open-hearth flange, fire-box or extra soft steel.

Rivets are made of open-hearth extra soft steel. Cross pipes connecting the steam and water drums of water-tube boilers, and cross boxes are of wrought or

cast steel when working pressure exceeds 160 lb. per sq. in. Mud drums of water tube boilers are of wrought or cast steel when working pressure exceeds 160 lb. per sq. in.

Pressure parts of superheaters, attached to boilers, or separately fired, must be of wrought or cast steel when the working pressure exceeds 50 lb. per sq. in.

Boiler and superheater mountings, such as nozzles, cross pipes, steam pipes, fittings, valves and their bonnets shall be of wrought or cast steel when exposed to steam which is superheated over 80 deg. F.

Water-leg and door-frame rings of vertical fire-tube boilers 36 in. or over in diameter, are of wrought or cast steel, or wrought iron.

47. Q.—What is the difference in quality of steel made by the basic, or open hearth process, and steel made by the Bessemer process?

A.—Bessemer steel as a rule contains a larger percentage of phosphorus, sulphur, and other impurities, while steel made by the open hearth basic process is of finer grain, and although its TS. may not be any higher than Bessemer steel, still it shows in tests a per cent of elongation that comes nearer to specifications, which is from 20 to 22 per cent in lengths of 8 in.

48. Q.—Why is Bessemer steel not used in the construction of marine boilers; and why do they use the open hearth steel in preference?

A.—Because the specifications for marine boilers require that open hearth steel be used in their construction. That is because open hearth steel being of finer grain and closer texture than Bessemer steel, withstands the very severe bending, and quenching tests which specimens of marine boiler material must undergo before being accepted.

Fusible Plugs

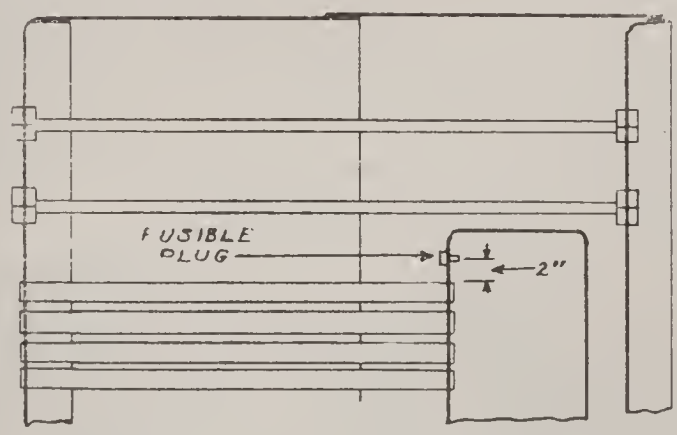
49. Q.—What is the best alloy for use in fusible plugs?

A.—The only filling recommended for fusible plugs is pure Banca tin. While in some instances an alloy has been used in place of the tin, to give a lower melting point, it has been found that this is not reliable as to its melting point, and the pure tin has a melting point low

enough, with a properly installed fusible plug, to take care of overheating before it becomes dangerous.

50. Q.—What is the proper location of the fusible plug for return tubular, vertical tubular, Scotch marine and locomotive type boilers?

A.—For a horizontal tubular, in the rear head, at a point 2 in. above the top row of tubes; for a vertical tubular, in an outside tube, at least one-third the length of the tube above the lower tube sheet. For the Scotch marine type, the plug is placed in the combustion cham-



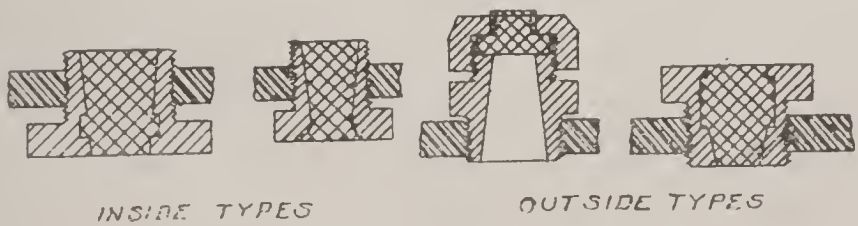
A50. LOCATION OF FUSIBLE FLUG IN DRY-BACK MARINE BOILER

ber; for the dry back type, in the rear head, not less than 2 in. above the upper row of tubes. For the locomotive type near the front of the top of the crown sheet.

51. Q.—Where is the fusible plug placed in a Heine boiler?

A.—In the front course of the steam drum, 6 in. above the bottom of the drum.

52. Q.—What is the proper place for the fusible plug in a Stirling boiler? Is it an inside or an outside plug?



A52. INSIDE AND OUTSIDE FUSIBLE PLUGS

A.—The fusible plug in Stirling boilers of the standard type should be placed in the front side of the middle drum, not less than 4 in. above the bottom; for super-

heater type, in the front drum not less than 6 in. above the bottom.

To have an inside or outside plug is simply a matter of preference, according to type of boilers. The outside plug would seem preferable on a Stirling boiler.

53. Q.—How can you get at the fusible plug in a Babcock & Wilcox boiler to replace it?

A.—The fusible plug in a B. & W. boiler is located in the steam drum, usually 6 in. above the bottom of the shell, and can generally be reached through the front doors of the boiler. In most settings of B. & W. boilers, there is a side clean-out door, which gives access to the tubes above the baffle plates.

54. Q.—Why is the fusible plug required to project 1 in. through the sheet or tube?

A.—The reason for the fusible plug projecting through the sheet is so that it may be heated up, if anything, sooner than the sheet proper; that is, if it projects into the hot gases, it will naturally heat to a temperature where the filling metal will melt before the sheet itself comes to that temperature. Also this prevents the end of the plug from being covered with scale or soot, if the boiler is kept at all clean.

55. Q.—What is the effect on the fire when a fusible plug melts?

A.—When a fusible plug is burned out, the immediate effect is a cooling of the furnace gases by the comparatively cool steam which rushes forth. In a horizontal tubular boiler, it will be seen that when the plug is burned, the effect as far as the draft is concerned, is to reduce it, the plug acting as a blower in a direction opposite to that of the draft.

In a vertical boiler, the burned plug would have no effect on the draft whatever. In the B. & W. and Stirling types, the blowing of a fusible plug would have no effect on the draft. The primary object of a fusible plug is to cool the furnace gases as a protection to the boiler against burning out.

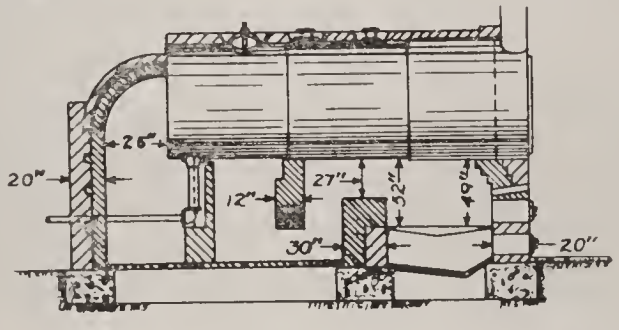
Boiler Settings

56. Q.—What should be the setting for three 72-in. by 18-ft. return tubular boilers as to furnace construction and height of boilers from grates, burning

mine run soft coal, with about 13,900 B.t.u.; have shaking grates, hand fired, 36-ft. grate surface and stack 120 ft. high by 54 in. in diameter. The boilers will be used two at a time to evaporate 60 tons water for distilled water ice.

A.—Looking over the data of the boilers which you have to set, we would recommend the following as dimensions for the furnace and setting:

The distance from the shell to the grate, at the rear, 52 in., set the front end of the grate about 3 in. higher. The distance from shell to bridge wall, 27 in.; from rear end of boiler to back wall, 26 in. Back and side walls, 20 in. thick, either made solid or built up with two 9-in. walls, having 2-in. space between filled with sand or



A56. SETTING FOR 72-IN. BY 18-FT. BOILER

some non-conducting material. Do not leave this as an air space. Bridge wall, 30 in. thick. The distance from shell to side walls, 4 in.

The furnace should, of course, be lined above the grate with fire-brick, preferably laid all with ends toward the furnace, as this makes a more durable and more easily repaired lining. It is well, also, to line the side walls in the combustion chamber, back of the bridge wall, with fire-brick, but below the top of the bridge wall the lining may be of ordinary hard-burned brick.

For the kind of fuel that you have, we would advise a mixing and combustion arch of fire-brick.

You will probably save something in radiation by setting two of the boilers in battery, and the third as a separate boiler. It is not desirable to set the third in battery, as this makes it impossible to get at the center boiler conveniently for repairs.

Your stack and grate surfaces seem to be all right, and the boilers have sufficient capacity to evaporate your water without any difficulty.

The material required for the setting of each boiler, assuming a full front setting, would be 2500 fire-brick.

16,750 common hard brick, 146 bu. of sand, 38 bu. of lime, and three barrels of fire-clay.

The rear brackets of the boilers will, of course, be supported on the side walls, and should have rollers under them to permit of expansion and contraction, and the back end of the boiler should be set 1 in. lower than the front end, so that it will drain toward the blowoff pipe. A brick pier should be built to protect the blowoff pipe from the impact of the hot gases.

57. Q.—What is the advantage of setting boilers high above the grates over a low setting?

A.—The distance between the grate and boiler shell should not be less than 28 in. when using anthracite coal, and not less than 36 in. for bituminous. The greater this distance is, the more complete will be the combustion, since the gas will have a better opportunity for combining with the air before coming in contact with the comparatively cool surface of the boiler shell. The vertical distance between the bridge wall and shell is usually from 10 to 12 in. for anthracite and 20 to 24 in. for bituminous.

58. Q.—What is the best method of supporting a h.r.t. boiler?

A.—For small boilers, the usual method of support is by means of lugs riveted to the side of the shell, and resting upon the brick setting of the boiler. For larger boilers, however, it is more advisable, and in some states required, that these boilers be provided with links, so that they may be readily supported by means of rods dropped from overhead steelwork, thereby taking the entire weight of the boiler off the brickwork. This is preferable and the safer of the two methods.

59. Q.—What are the advantages of suspending a steam boiler by hangers?

A.—A boiler suspended by hangers is free to expand and contract without affecting the brickwork of the setting. While a boiler properly supported on the brickwork with rollers, to take care of the motion, will work without cracking the setting, it is freer to move if supported by hangers; also, any settling of the brickwork does not tend to throw the boiler out of level, and replacing of the brickwork is more easily done.

60. Q.—What are two reasons for a boiler being lower at the back end than at the front?

A.—A boiler is generally set so that the rear end is lower than the front end in order to allow all sediment and sludge to settle at the rear and thus render it more easily removed by the blowoff. Also the water can be completely drained when laying up the boiler.

61. Q.—How thick should the side walls of a boiler setting be?

A.—The side walls are ordinarily made 16 to 18 in. thick, depending on the size of the boiler and the weight.

62. Q.—Has the use of air spaces in boiler setting walls been discontinued, and if so how is expansion of the inner wall taken care of?

A.—It has been determined by the Bureau of Mines that the air space in a brick wall is not of help, so far as the conduction of heat is concerned, and a solid brick wall is better. For a wall between boiler settings, we doubt if there is, except for a short time while the boilers are starting up, enough difference in temperature between the inside and outside of the wall to cause any expansion trouble; but if there is any doubt on this point, a 2 or 3-in. space might be left in the interior of the wall and filled solid with sand, which will prevent air circulation in this space.

63. Q.—What are the advantages of the Smith setting for a boiler?

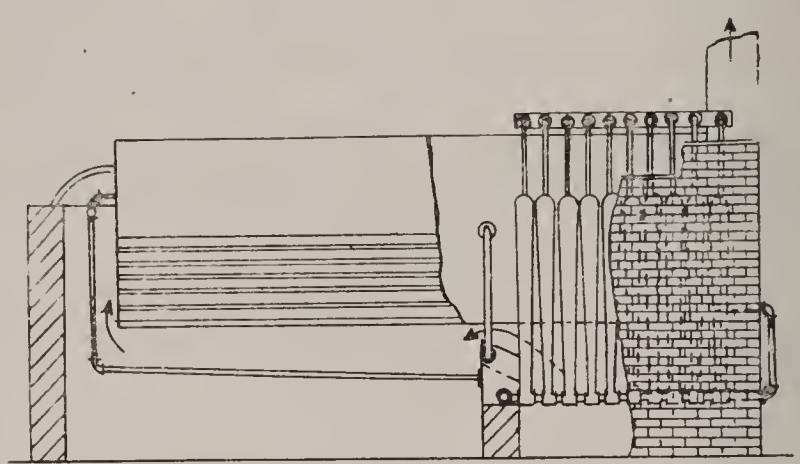
A.—The Smith setting for a boiler is an ordinary return tubular boiler with additional heating surface consisting of a hollow cast-iron bridge wall and a number of pipes run horizontally under and back of the boiler, connecting to short vertical tubes which, in turn, are screwed into a large horizontal pipe located back of the shell and connected to the shell. Also, two cast-iron pipes run along either side below the grate, connected with the water space in the shell, and from these cast-iron pipes a series of vertical cone-shaped castings, bulb-shaped at the top, are connected by small wrought-iron pipe to a steam header, and from there into the steam space of the main shell.

The object of the device is simply to utilize, in heating the water, some of the heat that otherwise might escape

through the brick setting, by arranging additional heating surface in the bridge wall and the brick sides of the boiler.

64. Q.—How is the equalizer made and used on a long heavy boiler, return tubular type, both on the side walls and in the outside suspension type of setting.

A.—The equalizer for a boiler with three supports on a side is simply some form, usually a turnbuckle or a nut on the end of the supporting bolt, so that all three



A63. SMITH BOILER SETTING

supports may be made to take as nearly as possible an equal part of the weight. Where but two supports are used, the boiler will naturally divide its weight between them, but for three supports, the two end supports might take all the load, and the middle one hang loose, or the middle one and one end might take all of the load, the other end not getting its proper share. By the use of a turnbuckle, preferably on the middle support, the length of this support can be changed so that it will take its proper part of the weight.

65. Q.—In connection with a boiler which is provided with 42 2½-in. tubes and a 36-in. grate the space between the rear head and the brick wall is 32 in., more than ordinarily allowed boilers of that size. There is a forced draft blower, and the stack is only 18 in. in diameter and 50 ft. high.

Would reducing the space between the boiler and the rear setting wall from what it now is to 11 in. bring about more satisfactory results?

A.—We do not see any advantage in reducing the space between the back end of the boiler and the brick wall. So long as this space is large enough to give free

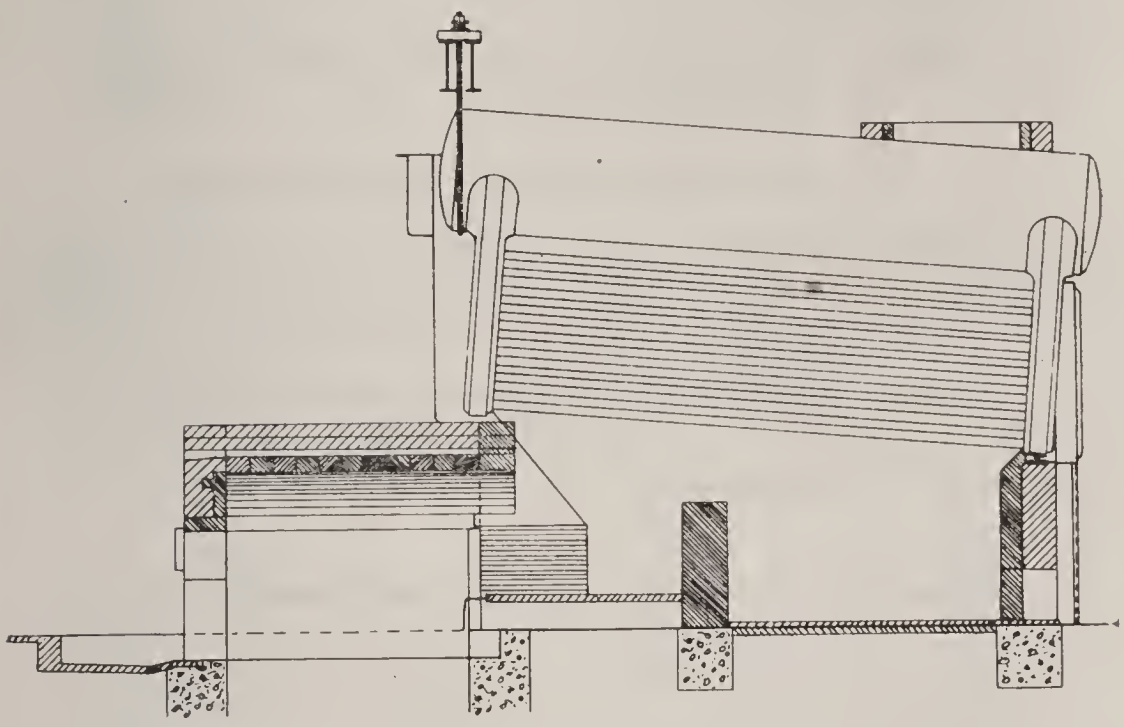
flow of the gases from the combustion chamber around into the tubes, extra space does no harm, except so far as it increases somewhat the radiation from the outside of the boiler setting.

This will not be of sufficient consequence to have any great effect on the operation of the boiler; in fact, if you are burning bituminous coal, the extra space would be an advantage in giving more time to complete the combustion.

66. Q.—What is a good and cheap covering for the tops of boilers?

A.—A good, cheap and efficient covering for tops of boilers, so that the heat losses from this source can be reduced to a minimum is: First cover top of boiler with coarse ashes from 6 to 8 in. thick; then cover this with a cement consisting of four wheelbarrows of coarse ashes, one bag Portland cement, one bag of asbestos cement thoroughly mixed dry, and wet. This makes a top sufficiently hard to walk on.

67. Q.—How is the Heine boiler supported?

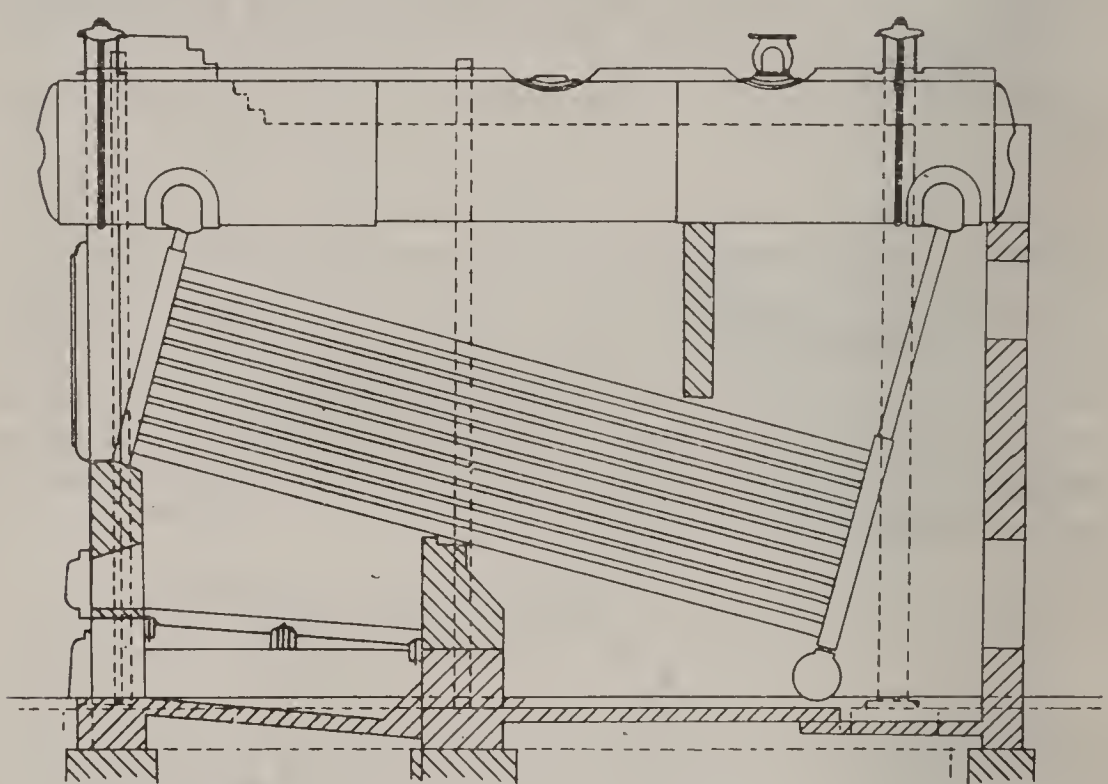


A67. SUPPORT FOR HEINE BOILER

A.—When set up ready for service, the Heine boiler inclines downward from front to rear, 1 in 12. The front end is supported on heavy cast-iron columns, the rear end resting on rollers which in turn bear on iron plates set in the top of the low and substantial brick wall which forms a part of the setting.

In case it is necessary to arrange the boiler to accommodate some stoking device, stout steel brackets are riveted to the water legs, in turn resting on special heavy cast-iron or steel columns, or an overhead support by which the boiler is suspended from above.

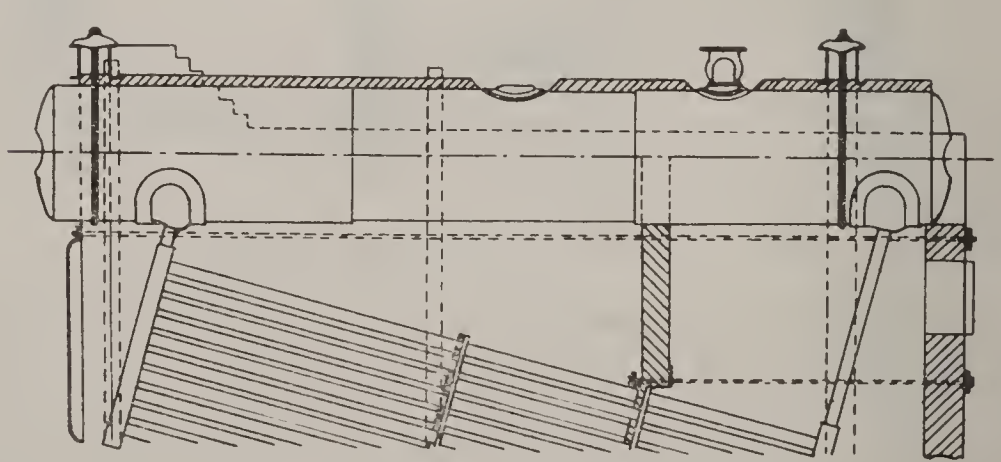
68. Q.—How is the Babcock & Wilcox boiler supported?



A68. SUPPORT OF BABCOCK & WILCOX BOILER

A.—The B. & W. boiler is suspended from I beams that rest on the side walls. Straps of steel are placed under the drums, and welded to these straps are bolts which extend up through plates resting on the I beams.

69. Q.—What holds the hanging bridge wall of a Babcock & Wilcox boiler?



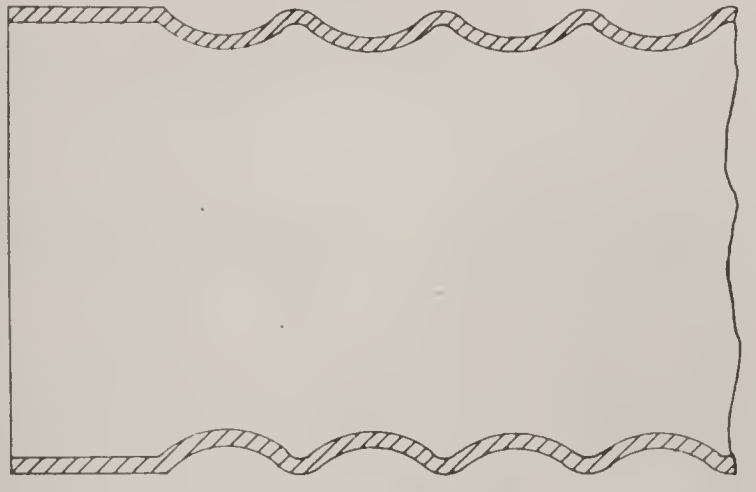
A69. SUPPORT OF B. & W. HANGING BRIDGE WALL

A.—The hanging bridge wall of the Babcock & Wilcox boiler is carried on an angle iron girder which is fitted with expansion sockets in the side wall.

Boiler Furnaces

70. Q.—What is a Morrison furnace?

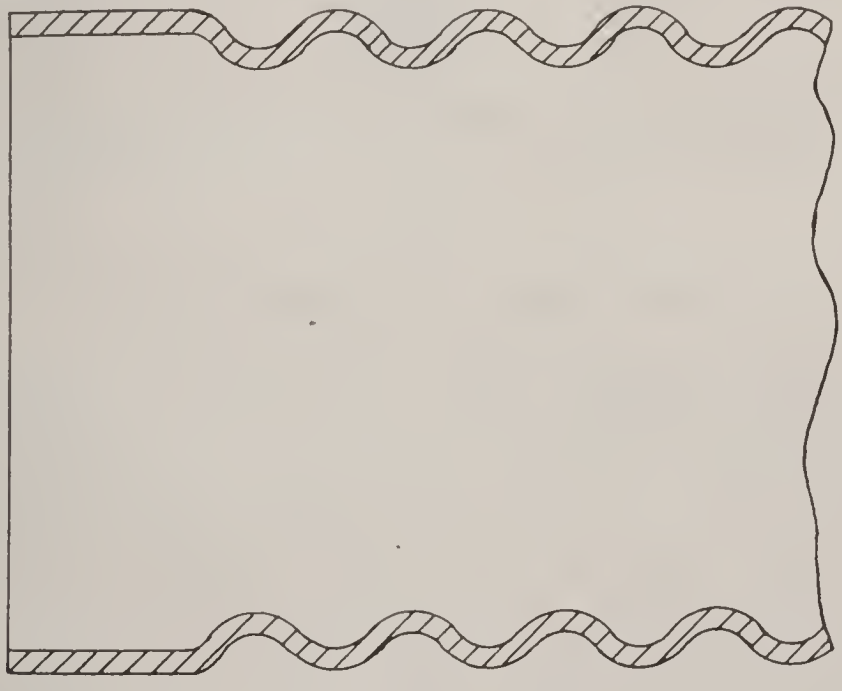
A.—The Morrison furnace is of the corrugated type; these corrugations help materially to strengthen the flue without the aid of stiffening rings. They can be purchased in various standard diameters, in lengths to suit, with one or both ends flanged.



A70. MORRISON FURNACE

71. Q.—What is the Fox furnace?

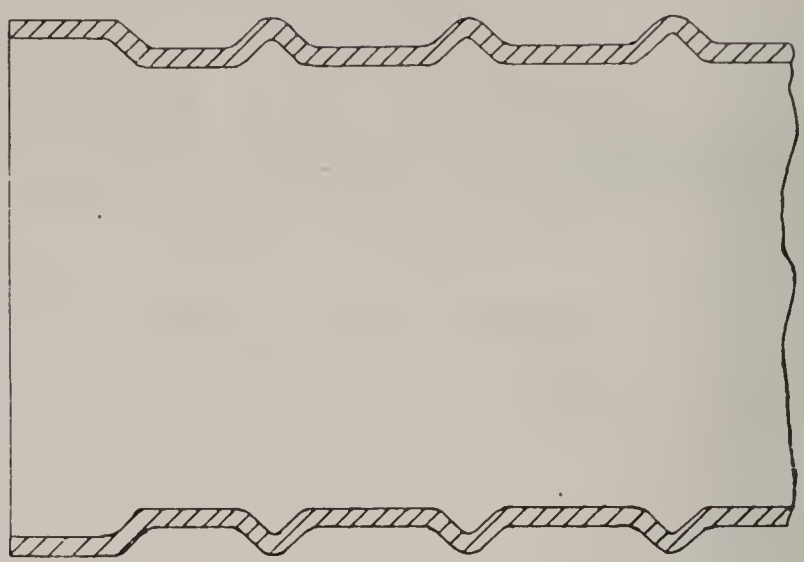
A.—This is made with corrugations semicircular, otherwise is like the Morrison. It is used principally in marine boilers.



A71. SECTION OF FOX FURNACE

72. Q.—What is the Purves furnace?

A.—This is ribbed at intervals, as shown in Fig. A72, the design depending on the diameter and pressure to be carried.

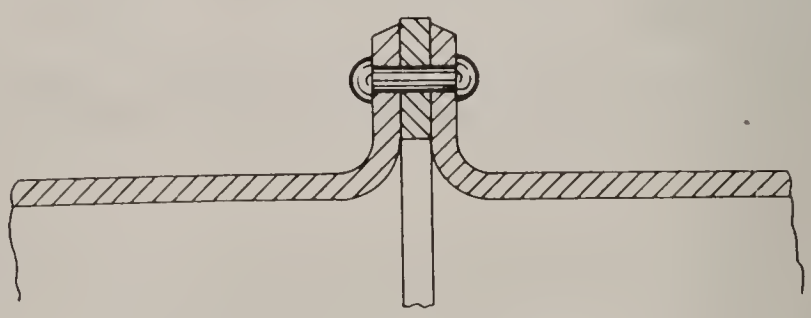


A72. PURVES RIBBED FLUE

73. Q.—What are the Leeds and Brown furnaces?

A.—The Leeds is similar to the Fox, and the Brown is practically the same as the Purves. The names are of the manufacturers instead of the designers.

74. Q.—What is the Adamson ring and where is it used?



A74. DETAIL OF ADAMSON RING

A.—The Adamson ring is a solid steel safety ring put between two sections of an internally fired furnace, to strengthen the furnace. The ends of the flue sheets are flanged out and are then riveted together, with the ring btween. The thickness of the ring should equal the thickness of the sheet, and the flanging of the flue sheets should be with a radius equal to two and a half times the thickness of the sheet. The detail is as shown.

75. Q.—How is the mean diameter of the Morrison furnace figured?

A.—In calculating the mean diameter of the Morrison furnace, the least inside diameter plus 2 in. may be taken as the mean diameter.

76. Q.—What is the allowable pressure on a Morrison flue 45 in. diam., of 3/8-in. plate?

A.—The maximum allowable working pressure, in pounds per square inch, on a Morrison furnace is equal to,

$$P = C \times t \div D$$

Where P is the maximum allowable working pressure, in pounds per square inch;

C = 15,600, the constant when corrugations are not less than 8 in. from center to center, and the radius of the outer corrugations is more than one-half of the suspension curve.

t = thickness, in inches, which should not be less than 5/16 in.

D = the diameter, in inches.

We have, therefore, by substituting the values you have given,

$$P = 15{,}600 \times 0.375 \div 45 = 130 \text{ lb.}$$

as the maximum allowable working pressure.

77. Q.—What is a "bridge-wall" in a steam boiler, and where is it located? Where is the "mud-drum" in a steam boiler, and what are its uses?

A.—The bridge wall is a wall with fire brick top extending across, under the boiler at the back end of the grate bars, from one side wall to the other, and up to within 12 to 20 in. of the bottom sheet.

The mud drum is a hollow cylinder constructed of the same kind of material as the boiler, but of much smaller dimensions. It is usually located underneath the rear end of the boiler, and connected to it by a throat of ample size to permit mud and other sediment to settle into the mud drum, whence it may be blown out through the blowoff pipe.

78. Q.—Which are the better for use in a 150-hp. water-tube boiler of the Keeler type, horizontal or vertical baffles?

A.—There is much dispute as to the question of vertical vs. horizontal baffles for boilers, some engineers preferring the one and some the other.

It seems to depend largely on the height of the boiler. If you have a high boiler, the vertical baffling will probably give the better result. If you have a low boiler, the horizontal baffling gives a better opportunity for gas travel. Especially if you have a low firebox, the use of a tile roof on the front end of the lower bank of tubes will tend to give better combustion than the vertical type.

Grates

79. Q.—What is the average allowance of grate surface to the horsepower?

A.—The Massachusetts Rules state that for boilers carrying 25 lb. pressure per square inch or less, gage, ⅔ sq. ft. of grate surface shall be allowed per horsepower; for boilers carrying over 25 lb., ⅓ sq. ft. of grate surface per horsepower. This difference is based on the fact that the low pressure, or heating, boiler usually has a low draft, and is run at a very low rate of combustion per square foot of grate per hour.

80. Q.—What determines the grate surface?

A.—The length and width of a grate depend entirely on the size of the boiler and its construction. The length should not be greater than that which can easily be fired by hand; usually 8 ft. is taken as the maximum length. The width depends entirely on the power which the boiler is to furnish, but the square feet of grate surface is commonly made such that coal can be burned at about 18 to 20 lb. per square foot per hour, and give the desired amount of steam.

81. Q.—How is the percentage of air space in a grate determined?

A.—In order to find the percentage of air space of a furnace grate, obtain the number of square feet of grate surface. Having this, it will be necessary to find the area of the openings in the grates, which may be done by finding the number of square inches in each opening and multiplying by the number of openings, dividing the result by 144. This will give you the area of openings in square feet. Dividing this last product by the

total grate area and multiplying by 100, will give you the percentage of air space.

82. Q.—What is the average ratio of grate surface to heating surface for anthracite coal? For bituminous coal?

A.—In hand-fired plants using anthracite coal under forced draft, 40 to 50 sq. ft. of heating surface to each square foot of grate has been found to give excellent results. Where bituminous coal is burned with natural draft there is variation in the ratio of heating surface to grate surface ranging from 50 to 78 with an average of 60. For stokers used with natural draft, conditions are much as with hand-fired furnaces, and excellent results are obtained with ratios of from 50 or 60 to 1, of projected area.

Proportions and Construction of Boilers

83. Q.—What is meant by the maximum capacity of a boiler?

A.—Maximum capacity is the greatest amount of power which the boiler can develop when it is burning all the coal it can, with the highest draft available. This may be rated either as horsepower, which is determined by dividing the total actual evaporation per hour by 30, or it may be determined by a per cent of the rated boiler capacity, which is found by dividing the actual horsepower developed by the nominal rated horsepower.

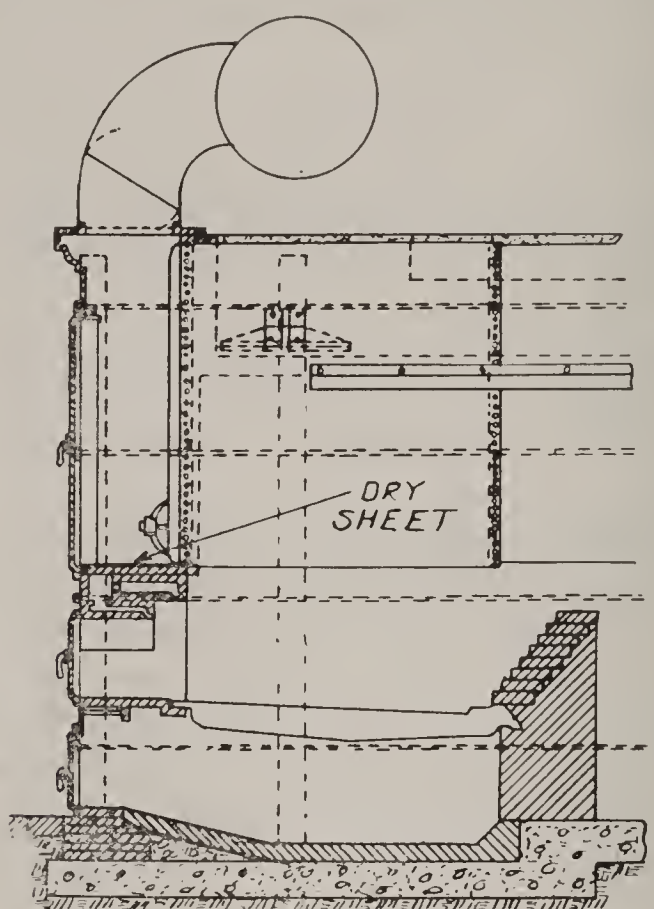
84. Q.—How is the heating surface of a boiler figured?

A.—To compute the heating surface of a return tubular boiler, we figure the external surface of all the tubes in the boiler, plus half the surface of the shell of the boiler, plus the area of the back head, and minus the area of the ends of the tubes which enter the tube head. The principle is to figure the surface of all parts of the boiler which have water on one side and hot gases on the other, that surface being taken which is next to the water. By this method, the heating surface of a B. & W. boiler is the inside surface of the tube plus half the surface of the steam drum. The heating surface for a vertical tubular boiler would be the sides of the furnace above the grate, plus the area of the lower tube sheet

minus the area of the ends of the tubes, plus the outside surface of the tubes up to the average water level.

85. Q.—What is the maximum thickness of boiler plate used on stationary boilers? Why are they not 1 in. thick?

A.—In general, stationary boilers have a shell thickness ranging from ¼ to ½ in., although boiler plates have been made in special instances with a thickness



A86. DRY SHEET OF BOILER

of 1½ in. When the thickness of the plate exceeds ½ in., the heat transmitting properties of the plate are reduced and boiler efficiency is lessened. Ordinarily a plate ½ in. or less in thickness provides sufficient strength to carry standard pressures and a 1-in. thickness would not only increase the cost of the boiler, but make it less efficient.

86. Q.—What part of a tubular boiler is known as the dry sheet?

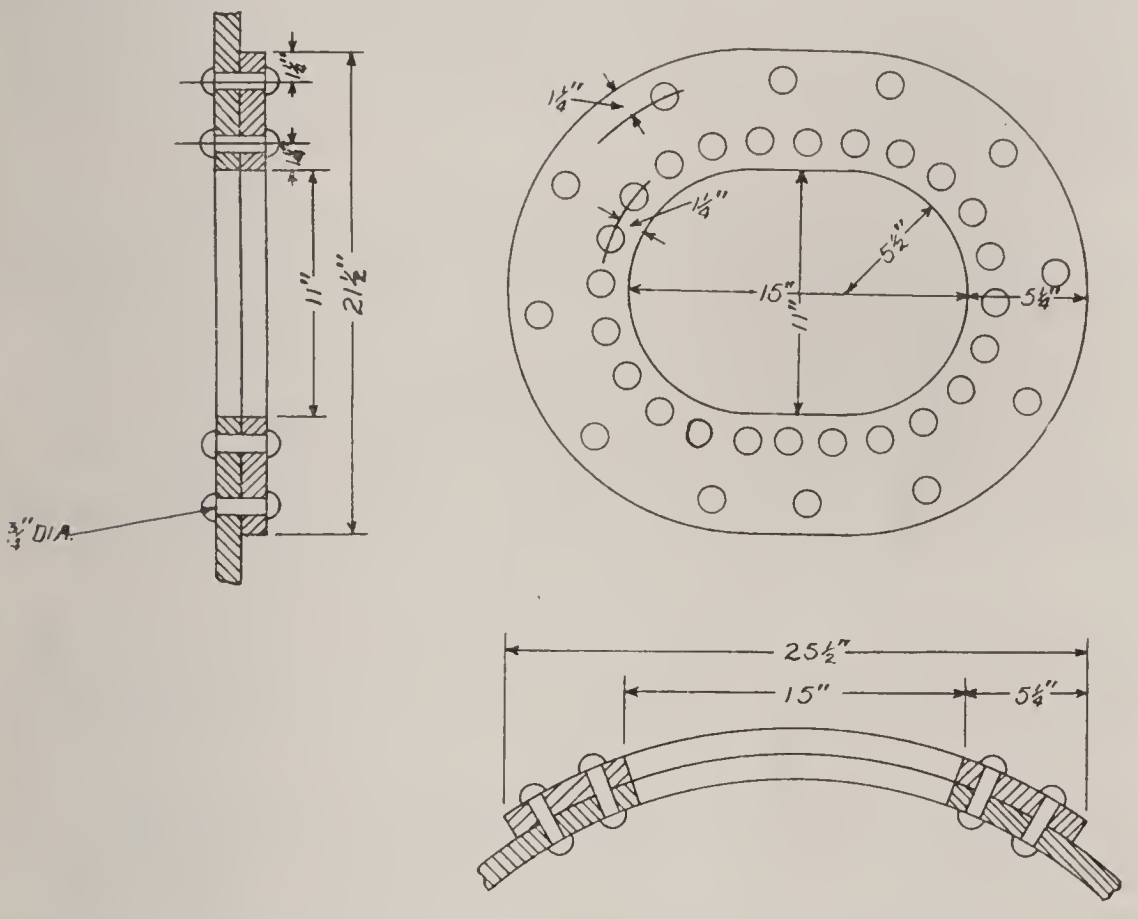
A.—In the flush front horizontal tubular boiler, the smoke box at the front end of the boiler is over the fire doors, and the part of the sheet which extends beyond the front head is called the "dry sheet." This is not

protected by water on the upper side, but has a firebrick lining on the under side to keep the heat of the furnace away from it.

87. Q.—What is the objection to having the front head of a h.r.t. boiler flanged inward the same as the rear head?

A.—If the front head were flanged inward, it would be almost impossible to get at the head to rivet it into the shell.

88. Q.—How would you construct the manhole frame for a 72-in. boiler?



A88. DETAILS OF MANHOLE FRAME

A.—The manhole frame should be of wrought or cast steel, with a net cross sectional area on a line parallel to the axis of the shell, not less than the cross sectional area of shell plate removed on the same line. For boilers over 48 in. diameter, the frame should be riveted to the shell or drum with two rows of rivets, having half as many rivets in the outer row as in the inner row. The strength of rivets in shear should not be less than the tensile strength of the shell plate removed on a line parallel to the axis of the shell through the center of the manhole. The manhole opening should not be less than 11 by 15 in. in size.

For a ½-in. shell with opening 11 by 15 in., the 15 in. being around the boiler as in Fig. A88, the shell section removed is 11 by ½ in. or 5.5 sq. in. If the frame is ¾ in. thick it will need a width of $5.5 \div 0.75 = 3\frac{2}{3}$ in. If ¾-in. rivets be used, we must allow for two ¾-in. holes, and $3\frac{2}{3} + 1\frac{1}{2} = 5\ 1/6$ in., or say 5.25 in. width of ring.

Tensile strength of plate removed $= 11 \times \frac{1}{2} \times 55{,}000 = 302{,}500$ lb., and if shearing strength of rivets $= 40{,}000$ lb. per sq. in., the area to be provided for shear $= 302{,}500 \div 40{,}000 = 7.56$ sq. in. Sectional area of a ¾-in. rivet is 0.442 sq. in., and $7.56 \div 0.442 = 17.1$ rivets, or say 18 rivets in each end, 6 on the outside row and 12 on the inside. If these be placed at 1¼-in. centers from the edges of the frame, the result will be as in Fig. A88.

89. Q.—Under what conditions can a boiler 60 in. in diameter withstand the same pressure as a boiler 48 in. in diameter, the shells of both being of ⅜-in. plates?

A.—The allowable steam pressure on any steam boiler may be found by the formula of answer 1.

The value of P, in the case under consideration, would be equal in both instances, that is, P is the same for a 48-in. boiler as for a 60-in. boiler, so that we have:

$$2\,S'\,t'\,e' \div f'\,D' = 2\,S\,t\,e \div f\,D,$$

and the left-hand side of this equation referring to the 48-in. boiler and the right-hand side to the 60-in. boiler.

Now, under the conditions stated in the problem, the thickness of the shell is in both cases the same, and we will assume that the tensile strength, or the value of S′ and S, is the same, as also the factor of safety, or f′ and f.

Cancelling these like values in the above equation, we have,

$$e' \div D' = e \div D$$

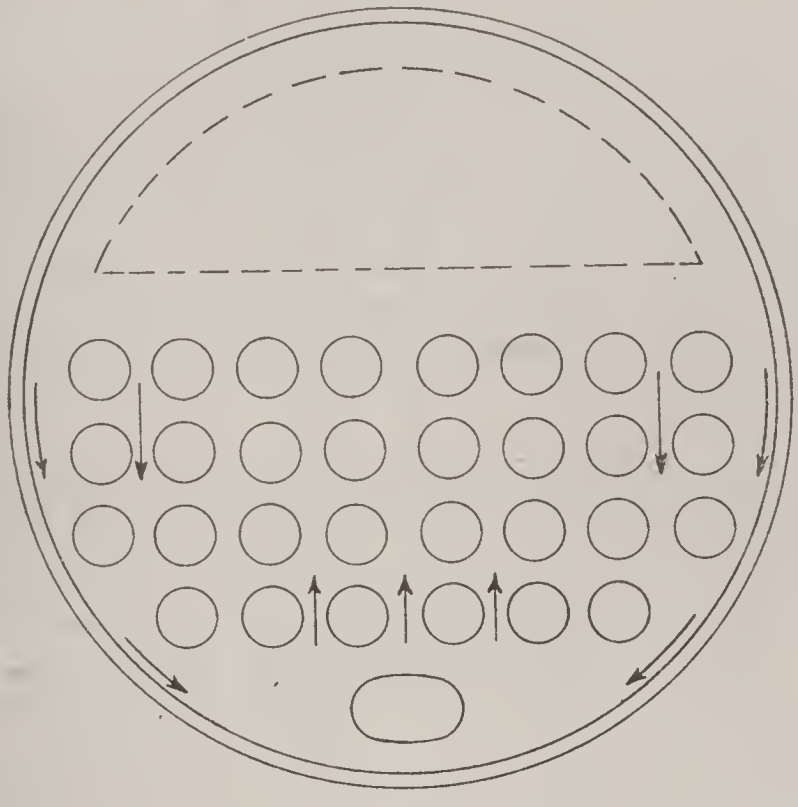
D′ is equal to 48 in., and D to 60 in., so that we have,

$$e' \div 48 = e \div 60$$

From this we see that the condition under which a boiler 60 in. in diameter can withstand the same pressure as a boiler 48 in. in diameter, is that the efficiencies of the joints vary directly as the diameters of the boilers.

90. Q.—How does the water circulate in a horizontal return tubular boiler?

A.—In a horizontal return tubular boiler, the hot gases from the furnace pass to the rear of the boiler and return to the front through the tubes inside the boiler. The water in this type of boiler circulates down near the shell and up around the outside of the tubes and by coming in contact with the hot tubes is heated and passes off as steam through the steam outlet.



A90. CIRCULATION OF WATER IN HORIZONTAL TUBULAR BOILER

The tubes in this type of boiler should be so arranged that the water can easily circulate over the heating surface of the tubes. This will be secured usually by arranging them in vertical lines with extra wide spaces at the middle and next to the shell. Some designers prefer, however, to arrange the tubes staggered vertically, claiming that in this way the water is made to pass over more of the tube surface, which is thereby rendered more efficient.

91. Q.—WHY ARE the shells of return tubular boilers made in three sections?

A.—It is not possible to get sheets wide enough to make a large return tubular boiler in a single section, and until recently it has not been possible to get them wide enough to make such a boiler in two sections. It is desirable to have the grain of the iron—that is, the

lengthwise of the sheet in rolling—run around the boiler, as it is stronger in this direction, and the greatest strength is needed in a direction around the boiler rather than lengthwise.

92. Q.—In 3-course return tubular boilers, why is the middle course placed inside the other two?

A.—This is to facilitate manufacture as much as anything, for by so doing, the two end sections are identically the same as are the two heads, and it is a much easier matter to work the sheets, and also to do the necessary laying out and drilling of the sheets.

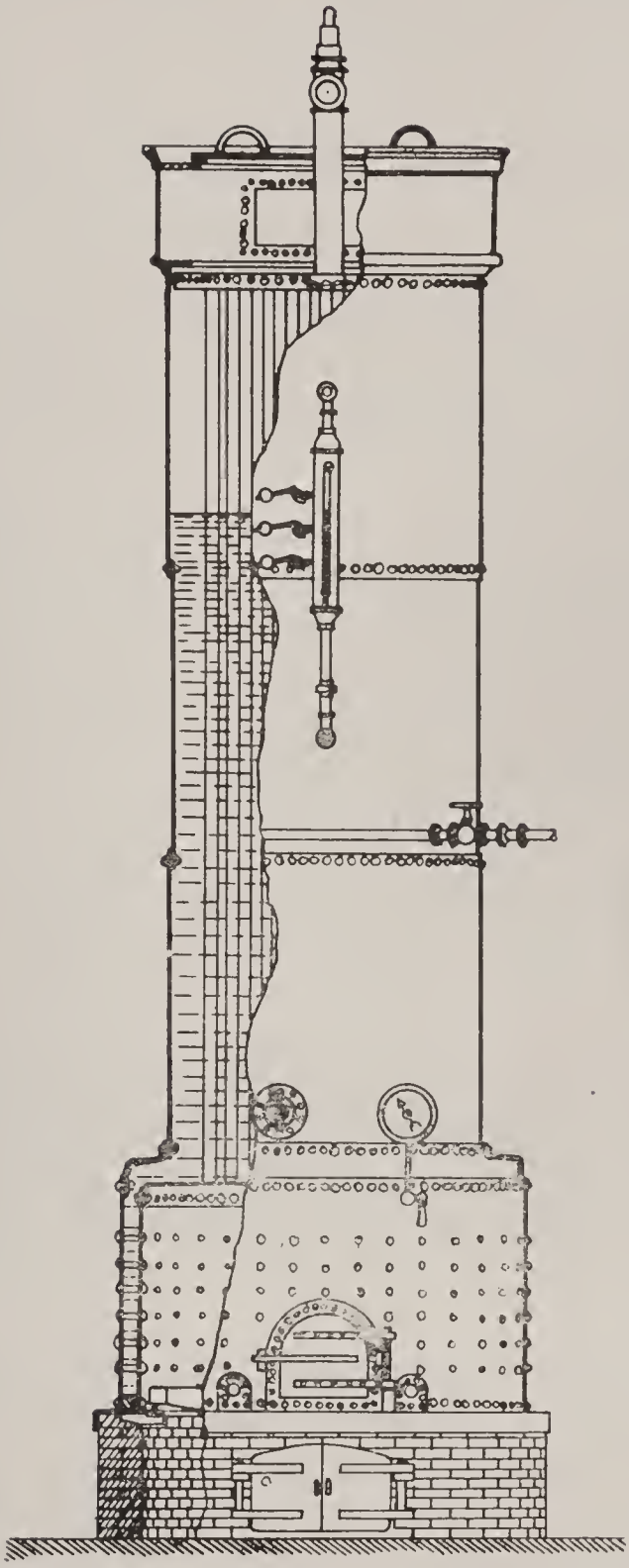
In service, the greatest heat from the furnace strikes the middle sheet and if there are any loose deposits in the boiler they would be more likely to lodge there and do harm if the middle sheet were outside of the two end ones so as to form a pocket between the circumferential joints.

93. Q.—Where is the bottom gage cock placed?

A.—There is no rigid location for the lower gage cock, but it is usual to place this in a horizontal tubular boiler, at least 3 in. above the top of the upper row of tubes. In a vertical tubular, it is placed at the point below which the water level should never go. If it is a submerged tube boiler, this is usually about 3 to 4 in. above the upper crown sheet. If the upper tube sheet is dry, it is then placed where the judgment of the designer of the boiler dictates, usually about one-fourth the length of the tube from the upper tube sheet.

94. Q.—How is the heating surface of a Manning boiler figured?

A.—The heating surface of a Manning boiler is equal to the internal area of the firebox, plus the area of the crown sheet minus the space taken up by the tubes, plus the internal area of the tubes up to the water line. The area of the firebox is equal to its diameter multiplied by 3.1416, times the height. The area of the crown sheet is equal to the diameter squared times 0.7854, minus the area taken up in the crown sheet by the ends of the tubes. To find the area of each tube, multiply the inside diameter by 3.1416, times the length of the tube up to the water line; this times the number of tubes will give the total area for the heating surface of the tubes. These various areas added will give the



A94 SECTION OF MANNING BOILER

total heating surface. This, however, does not include the superheating surface, which is that internal area of the tubes above the water line.

95. Q.—Which of the two water leg plates of a Manning boiler is the thicker?

A.—In the Manning boiler, the inner furnace sheet is invariably made thicker than the outer furnace sheet, owing to the fact that it is under compression. The outer furnace sheet is made thinner than it would be if it were not a staybolted cylinder, but the staybolts tend to prevent it from rupturing as it is being held by the staybolts, which, of course, are in tension.

96. Q.—Is there much circulation in the water leg of a Manning boiler?

A.—The circulation of water in the water leg of a Manning boiler has a tendency to be poor, as the volume of water is not large, and up and down currents tend to conflict.

97. Q.—What is the difference between the Manning and other vertical tubular boilers?

A.—The Manning boiler has a double reverse curve plate introduced between the water leg and the main shell, and the fire box is made much larger, in proportion, than in the ordinary vertical boiler, Fig. A94. This double curve, or ogee ring, is used for two purposes; first, to permit of the larger fire box, by increasing the diameter of the outside of the water leg; second, to take up the expansion strains in the boiler, permitting the tubes to lengthen and lift the top head of the boiler, this provision being taken care of by the flexibility in the curves.

98. Q.—Why does the feed pipe enter the water leg of a small vertical tubular boiler, while on a Manning it enters half way up the shell?

A.—In the case of a small vertical boiler, not as much attention is paid to circulation and efficiency as in a larger one. The best way to feed the Manning boiler, or any vertical boiler, is to put the water in at a point somewhat below the water level, pointing the discharge end of the feed pipe downward, so that the water will flow down with the course of circulation, and then rise in the center of the boiler. In many Manning

boilers, the feed is discharged into the water leg, particularly when it is heated before feeding in. The small vertical boiler has the water fed in at the bottom, as this being the coolest part of the boiler, there will be the least harm done here of anywhere, and the expense is not desired of putting in a length of internal feed pipe.

99. Q.—What is the object of a blowoff, and how often would you examine it?

A.—A blowoff pipe is for the purpose of relieving a boiler of all, or a portion of its water; and also for carrying off the mud and sediment from the boiler. The blowoff pipe should be examined at least once a day if not more often, depending on the amount of deposit from the water.

100. Q.—What is the best pipe for blow-off lines?

A.—Nothing less than extra heavy pipe and fittings should be used as blow-off pipe of a boiler, and it should not be larger than 2½-in. If the boiler is of the water-tube type, where it is possible or necessary to do so, two blow-off connections should be provided.

101. Q.—What size blow-off tank is needed for a battery of three 100-hp. boilers? Is brick or concrete suitable?

A.—It is usual to provide, in a blow-off tank or basin, about 1/24 the volume of the boilers to be blown off into it simultaneously. In case of three boilers, it is likely that all of them might not be blowing at the same time, or before the tank could be emptied.

The volume of a 100-hp. boiler would be about 320 cu. ft., or for three, 960 cu. ft., and 1/24 of this would be 40 cu. ft. as the capacity desirable for a basin. This is on the supposition that the basin will be emptied into a sewer or other drainage after the water has a chance to cool and give up its steam.

There would be no objection to the use of concrete, provided a good coat of neat cement is put on the inside of the tank. This should be at least ½ in. thick. Brick would hardly be desirable, although it would be entirely possible to build the tank of brick, laying it up with Portland cement mortar, and then to give this a coating on the inside of neat cement ½ in. thick.

102. Q.—What are the causes of leaky tubes?

A.—Internally; sediment, grease, scale on tube ends at heads, or on heads at tube ends; corrosion from feed water or from acid in boiler compound.

Externally; cold air from cracks in setting; from intermittent firing, from tube blower, from brickwork in combustion chamber causing unequal distribution of heat.

103. Q.—How should a bagged boiler be repaired?

A.—To reduce a bag in a boiler, if it is of large size but not very deep it will not be necessary to drill the bag in the center, but if it extend some distance along the general surface of the sheet, it is necessary to drill a hole in the center of the bag in order to give the metal a chance to upset, as otherwise it would buckle and at best make but a poor job. When the bag is driven back in place, ream the hole out and put in a rivet well driven up and tight to the sheet.

The method of heating the sheet before driving back the bag, depends much on the facilities at hand to do it with, but as a small hand forge can be secured in nearly every vicinity, we will describe the method of using this first.

First, clean the shell and interior of the furnace thoroughly so that the pounding on the shell will not dislodge the soot to the discomfort or inconvenience of the workers, as the loss of but a few seconds sometimes will change an otherwise good job into a very poor one, and every second counts. Remove the grates.

Then erect the forge under the bag in such a manner that it can be quickly removed completely out of the way to permit the use of hammers or sledges. Build a charcoal or coke fire in the forge and try to heat the bag uniformly all over and at the same time to protect the sheet around it as much as possible, so that it will be solid and not affected by the force of the blows.

Get all the tools that are required right at hand, so that they can be taken up without the loss of a second of time. Sledges of about 8 to 10 pounds with a handle short enough to make handling easy and sure are best, also a "flatter," and a short metal straight edge, such as a common square.

When the bag has attained the proper temperature to work easily, quickly move the forge out of the way and then two good men take the sledges and carefully drive the bag back, taking care that it is not dented unnecessarily. When nearly in place, use the "flatter" and straight edge till the surface of the bag conforms to the remainder of the shell.

Do not try to work the metal if it gets too cold. If necessary, set up the forge again and take another heat. In some vicinities it is possible to get a large gas or oil blow torch for the purpose of heating the bag and then possibly, if convenient to do so, when once heated the flame could be inserted in the front manhole of the boiler so that it impinges on the inside of the bag and thus keep it to a good working heat.

Another method of heating, if a supply of molten iron is available, is to fill the bag on the inside with the iron throwing the end of an old piece of chain into the bag with which to pull the solidified metal out through the manhole as soon as the sheet is hot enough. Great care must be exercised with this method to see that there is no dampness in the bag, and also not to get a chunk of metal so large that it cannot be withdrawn through the manhole.

104. Q.—What is the difference between a bag and a blister?

A.—Bagging of a boiler plate is caused by overheating, generally due to scale or other deposits accumulating to such an extent that water is prevented from coming in contact with the plate. Under the influence of heat the plate then becomes soft and easily yields to the pressure acting upon it, thereby forming a bag.

A blistered boiler plate is one having, through imperfect material, blisters on its surface. This is caused by the layers of the plate separating, so that the outer layer being abnormally expanded by the heat, forms a blister.

105. Q.—With a bag on a boiler, how can the true thickness of the metal at this point be found?

A.—Knowing the thickness of the plate and the size and depth of the bulge, the thickness at the lowest point might be approximated, but the only sure way is to drill a hole in the lowest point of bulge and measure the plate.

106. Q.—How can the tensile strength of the metal at the bulge be found?

A.—The tensile strength will be found stamped on plates, and will be affected injuriously in proportion to the degree of heat to which plate was subjected at the time of bulging. If burned, the tensile strength will be destroyed.

107. Q.—In a boiler which leaks at the center seam just above the brick work, would it be dangerous to allow this condition to continue for some time? How would you advise repairing?

A.—Usually leakage in the center seam above the bridge wall is due to a deposit of scale at this point. If that is the case, it is dangerous to let it go, as the accumulation of scale is likely to produce overheating, and that is probably what has caused the leaks.

The boiler should be emptied and examined carefully to see whether there is scale deposit at this point. If there is, it must be removed, and the joint examined for any possible cracks. If the metal is in good condition and shows no sign of overheating, the leaks, unless bad, can be taken care of by recalking the seam. If the leaks are bad, it will probably be necessary to cut out the rivets and re-rivet the seam.

Of course, if there are cracks in the plate, it will be necessary to have extensive repairs, as this is not a good place to put a patch.

108. Q.—Name the different kinds of patches used on a steam boiler. Which is the more reliable? And why?

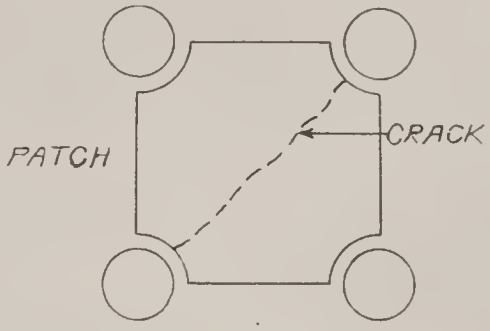
A.—Riveted patch, and bolt or soft patch. The former is the more reliable, for the reason that it is joined to the main sheet by rivets in the same manner that a lap joint is made.

109. Q.—Would it be advisable to put a soft patch on a Scotch marine boiler? Especially on the tube sheet when there is a crack between tubes for carrying 150 lb. pressure?

A.—Unless it is absolutely necessary, it is not advisable to put a soft patch on any kind of a boiler, and the only place where it would be permissible to use this patch would be on a part not exposed to the direct heat

of the gases. In the case cited in your letter, it would not do to apply a patch such as you have shown, unless this was absolutely necessary, and it should be replaced as soon as possible.

A crack in the tube sheet, such as shown, is quite serious; even though it involves considerable expense, it is advisable to remove this sheet and replace it with a new one, and not assume any risk whatever. A soft patch may be placed on a boiler carrying 150 lb. pressure, but only under the above conditions.



A109. CRACK IN TUBE SHEET AND PATCH

110. Q.—WHAT EFFECT will accumulation of soot in boiler tubes have?

A.—An accumulation of soot around or in boiler tubes will seriously interfere with the proper transfer of the heat from the gases to the water in the boiler, and will have a tendency to reduce the efficiency of the boiler.

111. Q.—IN CLEANING boiler tubes, which is the better—a dry brush or a steam jet?

A.—Both have their advantages and their disadvantages. No matter how careful one may be to drain the steam connections to the steam jet, there may be a small amount of moisture in the steam which will cause a certain amount of the soot to adhere to the tubes and to become hardened. With a tubular boiler, while the tubes ought to be blown about every 10 hr. of operation, yet with the recent devices for insertion in the combustion chamber, they can be blown at will, with very little time.

But the writer believes that no matter what type of steam jet be used, a good scraper ought to be used in addition at least once a week or so, to remove hard scale. The writer once took charge where the fuel was

rock elm shavings, and found it pretty "hard sledding" for a few days, but on investigating, found the tubes filled with a hard scale which required an entire day with a scraper to remove (76 tubes). After this the tubes were blown daily as usual, and scraped weekly. The fuel saving was remarkable.

The water-tube boiler, of whatever type, presents a problem that defies any other method of cleaning tubes but by blowing, either by steam or air. Some of the horizontal water-tube boilers are designed in such a manner that it is possible to remove the crust from the fire surface by means of a small chain wrapped once around the tube and drawn back and forth, but this can be done only on the tubes most accessible to the operator.

112. Q.—How could you tell on internal inspection that a boiler had had oil in it?

A.—Oil on the inside of a boiler may be recognized by the presence of a number of small globules of water on the surface of the shell.

113. Q.—What is the best method for removing oil from boilers?

A.—When possible to do so, after emptying the boiler, pour several pails of soda ash solution into it. Fill the boiler with water, and fire gently for several hours, not allowing the pressure to exceed 15 lb. From time to time, let out part of the water and replace with fresh water. Continue for about 12 hours. Then cool boiler, empty it, and rinse with fresh water.

114. Q.—What precautions are needed in putting a new tube in the bottom row on a B. & W. boiler? How would you get the old one out? What kind of a cutter is used?

A.—Ordinarily the method used in cutting out a tube is as follows: First, two slots about ¾ in. apart are cut in the bottom of the tube with a flat ripping chisel to the inside face of the seat. This narrow strip is then bent up with what is known as an oyster knife. A round nose chisel is then used for cutting the tube inside of the seat for a distance of 4 to 6 in. The slotted portion of the tube is then turned in and the diameter of that portion originally expanded to make a seat is decreased so that the tube may be drawn through the

tube hole. In cutting out a tube, great care should be exercised not to injure the seat, that is, not to cut a groove or slot across the seat. All cuts should be made clear of the seat.

115. Q.—What should be done for leaking hand holes in a Babcock & Wilcox boiler?

A.—The B. & W. boiler of late years uses a hand-hole plate which has a ground joint on the outside and the crab on the inside of the sheet. With this method the entire strain of the pressure as applied to the inside of the plate comes on the bolt which holds the plate on. In case of leakage there is no recourse but cooling the boiler off and regrinding the plate. Drawing up on the bolt when under pressure is bad practice and should be discouraged.

116. Q.—How can a cracked header in a Babcock & Wilcox boiler be repaired?

A.—The only way of repairing a boiler with a cracked header is to remove the header and replace it with a new, which can be done without sacrificing the tubes.

117. Q.—For bent boiler tubes of the Stirling type, re-ended about 6 in. on each end by the acetylene process; is it as good as charcoal weld? If not, why? Why will not boiler insurance companies carry and pass on them? Why will they pass on a fire-tube and not on water-tube boilers?

A.—In regard to re-ending tubes in a Stirling type boiler by the acetylene process, it is doubtful if an insurance company will write a policy on a boiler with the tubes so treated.

The principal objection raised by insurance companies to welded construction or repairs in steam boilers is that the efficiency of a weld, at best, must be assumed. In fact, there are so many conditions that tend to make the actual strength of a weld more or less of a guess, that the insurance companies have in many cases forbidden their inspectors even to consider as a risk boilers having welded parts, especially where the welded part is in tension—as would be the case with tubes in a Stirling boiler.

This will be better understood when it is explained that practically all policies written on water-tube boilers

include tube failures. The reason for this is that the loss of a tube in a water-tube boiler means dropping that boiler from service until repairs are made, which is not the case with fire-tube boilers, many of which run for months with plugged tubes.

It would be well to keep in mind also the fact that water tubes are subjected to an internal or bursting pressure while fire tubes are subjected to a collapsing pressure—and this in itself is a serious objection to welds in water tubes.

The writer rather favors charcoal welding, especially where the article to be welded is large in size, as this method of welding imparts a more even temperature to the article to be welded, which is an important consideration in all welding. In fact, the statement is made that the charcoal process practically eliminates the human factor in welding, which is really the direct answer to good or poor welds.

A still further objection raised by some insurance companies against welded tubes in water-tube boilers is that once the tube is installed it is not accessible for inspection, and no matter how thorough the inspector may be, he has absolutely no means of ascertaining the exact condition of the tube welds except on the outer surface of the tubes. Some companies are said to have instructed their inspectors to insist on a yearly hydrostatic test on boilers with welded parts; but to the writer's mind the value of even this test is questionable, as the conditions under which such test is applied and the conditions under which a boiler operates are very different.

To sum the matter up, the installation of tubes re-ended by the acetylene process, is not recommended if it is desired to have the boilers insured.

118. Q.—For two 100-hp. boilers connected to the same steam header, how can more work be gotten from one than from the other?

A.—In order to obtain more work from one of the boilers than from the other, close the ash pit doors and dampers of the boiler which you desire to have do the least work, i.e., reduce the draft, and the fuel burned, and in this way you can proportion the load between the two units in whatever way you desire.

119. Q.—With water glass clear, the gage cocks

show a scant two gages of water while the glass shows 2½ gages. What is wrong?

A.—As you know, the water level indicated by the gage cocks is but approximate as compared with that indicated by the glass, and it may be due to this discrepancy that the levels in the two appear to be different. Again, due to being in direct contact with the external air, more or less of the steam in the water glass is continually condensing, with the result that the glass may show a water level a trifle higher than that of the boiler or water column. This condensation is, however, slight and hardly of enough consequence to cause any great difference in the indication of the water level.

120. Q.—What is the proper height at which to keep the water level? How does the height affect the coal consumption?

A.—Many engineers firmly believe it more economical in coal consumption to carry the water in their boilers quite high. The theory advanced by them is that with an extra supply of water in the boilers, it is unnecessary to introduce comparatively cold feed during the time the boiler may be most heavily drawn upon for steam, thus allowing all available heat to be used for steam generating purposes rather than for heating the feed water.

With the water in the boiler quite near the bottom of the gage glass and a load coming on, much of this water will be evaporated, necessitating the rapid introduction of comparatively cold feed water. This requires the coal to supply heat not only for converting that water already in the boiler into steam, but also for raising the temperature of the feed water then introduced to that of the water already in the boiler. As a consequence, more coal is naturally required.

If water is carried quite high, a heavy load can be carried for some little time before any more water must be fed to the boiler, all the heat of the fuel being available for steam generating purposes only. Then, as the load gradually decreases, advantage may be taken of the available heat in the heavy fires by restoring the water level. In this way—namely, that of feeding the boiler light during heavy loads and heavy while the load is dropping off—a maximum amount of water may be evaporated with a minimum amount of coal.

Of course, if the load does not vary and just sufficient feed is introduced to make up for the water being converted into steam, it is immaterial whether the water in the boiler is near the bottom of the gage glass or near the top.

121. Q.—If you have a boiler with an inch of oil in the gage glass, and you wished to blow down the boiler, what would you do before blowing the boiler down?

A.—With oil floating on the surface of the water, blowing down the boiler through the blowoff will serve simply to let the oil wipe down along the inside surface of the shell and over the tubes, coating them with a layer of oil and leaving the greater part of the oil inside the boiler. Oil floats on the surface of the water, and as the water falls, the oil will be deposited on tubes and shell and cling there.

The best thing to do would be to put in sufficient soda ash or lye to cut the oil and make soap of it. This would be soluble in the water and would blow down through the blowoff. If this is not readily possible, the boiler can be blown down and then washed out thoroughly with hot water, which will carry off most of the oil; but this washing must be done with a strong stream of hot water from a hose nozzle, so as to scour the oil off the surfaces.

Oil in the gage glass may not, however, mean the same amount of oil in the boiler, and it would be well, if possible, to take the boiler out of service, cool down, take off the manhole cover and make an internal examination to see just how much oil is in the boiler. If there is any amount, find where it comes from and take steps to prevent further feeding of oil into the boiler.

122. Q.—Which is more economical firing a boiler under light or heavy load?

A.—Usually a boiler under light load will show a slightly better economy than under heavy load, but with a properly designed boiler, having clean surfaces, the difference in efficiency will be small.

123. Q.—What precaution would you take while blowing down a boiler with a good hot fire under it?

A.—No boiler should be entirely blown down or emptied with a hot fire under it. If the blowing down

is, however, only for the purpose of removing sludge or sediment, the blowoff valve should be opened for but a short time at frequent intervals. A good method to follow is to open the valve gradually until wide open and then at once slowly close it again. If there is a valve and a straightway cock, open the cock first, then the valve. Close in reverse order.

124. Q.—Would you blow down a boiler with pressure on it if letting down to wash out?

A.—No. Allow pressure to fall to practically zero before blowing down entirely. It is, of course, permissible to open the safety valve occasionally to reduce the pressure gradually. If, however, there is no particular hurry, the best method is to leave the boiler alone, allowing the pressure to fall as it will; and in this way, let the shell cool off without being subject to any severe strains.

125. Q.—What is a good method to test for leaking blowoff on a battery of boilers?

A.—A test cock in the bottom of the blow-off pipe near each blow-off valve will, when opened, give warning of leakage. If there is no room for such a cock, the joint between the branch and the main may be opened and the leakage, if any, from the boiler end noted.

126. Q.—When a blowoff is plugged with scale, can it be cleared by blowing pressure from the next boiler, if all are connected?

A.—A clogged blowoff may or may not be cleared by attempting to blow through from another boiler depending upon the nature of the obstruction. This involves closing the end of the blow-off and turning full boiler pressure into it, which is always a proceeding of doubtful wisdom. Generally blow-off branches are equipped with two valves or a valve and a cock. Should the obstruction be outside of the valve next to the shell, it may readily be removed by disconnecting that section of the branch between this valve and the blowoff main. Should the obstruction be between the shell connection and the first valve and it cannot be blown out, shut down the boiler at the first opportunity after which the entire blowoff branch may be taken down and the obstruction removed.

127. Q.—How DO you clean out the mud drum of a Stirling boiler?

A.—This should be blown out at regular intervals, which must be determined by close observation in each case. Occasionally some mud adheres to the rear bank of tubes, and this should be regularly removed by washing down with water applied under a good pressure through a hose terminating in a rose-shaped nozzle.

128. Q.—How WOULD you cool a horizontal return-tubular boiler in the shortest possible time?

A.—To prepare a boiler for cleaning, the fire should be allowed to burn as low as possible and then pulled out of the furnace, the doors being left slightly open and the damper wide open in order that the walls may gradually cool. On account of the injurious effects on the plates and the seams of the shells and the brickwork of the setting, it is just as bad to cool a boiler off suddenly as to fire it up too quickly. After the boiler has become sufficiently cool and no more pressure is indicated by the gage, the blowoff valve may be opened and the water allowed to run out. If operating conditions possibly allow doing so, a boiler should be left for 18 to 36 hr., after being cut off from the line, after which it may be emptied, opened and entered.

129. Q.—WE HAVE an 80-hp. boiler that has been used for furnishing steam for a 35-hp. engine and for laundry machines and dry rooms. We have now installed motor drive for all machinery, and only use the steam for heating the laundry machines and washers. I believe that a 50-hp. boiler would more than meet out requirements. Will reducing the grate area by bricking up side walls and bridge walls from 5 to 3½ ft. cut down the horsepower as well as coal consumption?

A.—There is no question but what you can reduce the horsepower of your boiler and the coal consumption by bricking up the side walls and bridge walls so as to reduce the grate area. You will want to be sure, of course, that your boiler setting is tight, as with the reduced grate area there will be a much greater tendency for air to leak into the setting and around cleanout doors and similar openings, and this will cut down efficiency.

The cutting down of the grate should all be done at the sides, not reducing the length at all, as this would

tend to hurt your conditions for good combustion, particularly with the coal that we are getting these days.

130. Q.—What are three methods of hand firing boilers?

A.—Three methods of hand firing are: The coking method, in which green coal is fed onto the dead plate in front and pushed back after it is partially coked; alternate firing, in which first one side of the fire is covered and then the other; and spread firing, in which coal is thrown onto the entire surface of the grate at once.

131. Q.—What is the best result—pounds of water evaporated per pound of coal—that can be obtained by hand firing?

A.—With a good grade of coal and careful hand firing, it is possible to obtain, with a good furnace and boiler, an evaporation of from 8 to 9 lb. of water per pound of coal.

132. Q.—What per cent saving will a stoker give over hand firing?

A.—Just as good boiler economy can be secured with the best hand firing as with a stoker. The principal disadvantage in hand firing arises from the opening of fire doors, which, of course, is eliminated with the stoker.

133. Q.—How can the draft over the fire in a horizontal water-tube boiler be found?

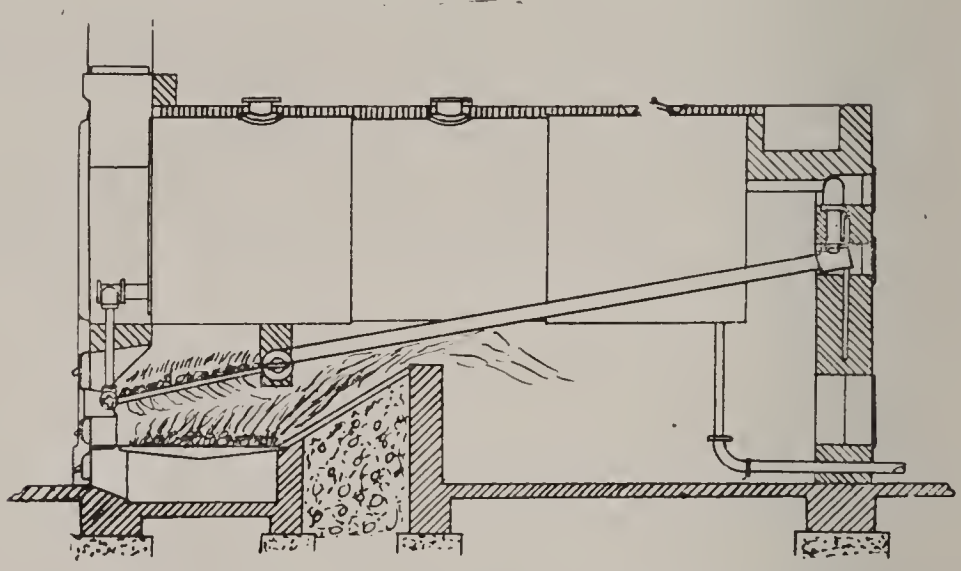
A.—In order to find the draft over the fire of such a boiler, insert a ½-in. pipe through the wall of the boiler setting. At the end of the pipe in the furnace, connect an ell having an extension of the same size pipe about 12 or 18 in. long. This extension should be parallel to the path of the gases, and the opening of it should be placed away from the fire doors. Then, by means of a piece of rubber tubing, connect the end protruding from the boiler wall to a draft gage. The reading of the draft gage will give the draft over the fire in inches.

134. Q.—What are natural, forced and induced draft?

A.—Natural draft is caused by the passing of air upward through the grate bars and into the stack. In forced draft the air is driven under the furnace by fans. With induced draft the gases are drawn from the boiler by means of fans which discharge into a short stack.

135. Q.—In a battery of four boilers, which has the best draft, and why?

A.—With a battery of four boilers, the best draft will naturally be on the boiler nearest the chimney, as the draft will be reduced on the other boilers by the amount necessary to overcome the friction of the smoke



A136. SECTION OF HAWLEY DOWN-DRAFT FURNACE

pipe. Frequently it is attempted to overcome this somewhat by using a tapered smoke pope which is larger near the chimney than at the far end, thus tending to reduce the friction of the gases in the pipe.

136. Q.—What is the Hawley down-draft furnace? Kindly give some hints relative to its operation.

A.—In the Hawley down-draft furnace, there are two separate grates, one above the other, the upper one being formed of parallel water tubes connected with the water space of the boiler through steel headers or drums, in such a manner as to ensure a positive circulation. Fuel is supplied to the upper grate—the lower one, formed of common bars, being fed by the partly-consumed fuel falling from the upper grate.

Air for combustion enters the upper fire door which is kept open, and passes first through the bed of green fuel on the upper grate, and then over the incandescent

fuel on the lower grate. A strong draft is required, due to the relatively small upper grate area and the correspondingly high rate of combustion.

Lump coal gives better results than the smaller sizes, as the latter are apt to fall through the upper grate before being even partially consumed; and when such is the case, efficient results cannot be obtained.

If carefully manipulated, this furnace, with fire-tile tubes, gives high boiler efficiency and smokeless combustion, but its overload capacity is limited. Without the fire tiling, smokeless combustion is possible only at light loads.

The theory of this furnace is, that the gases distilled from the green coal are mixed with air, ignited and burned while being passed through the underlying incandescent fuel. After the coal is coked, it is expected to drop through the grates upon the second grate. Combustion of the coke or fixed carbon is completed upon this second grate.

It can be seen that such a furnace has somewhat narrow limitations with respect to coal. If, for instance, the coal is too fine, it will fall through upon the second grate before coking is completed, and there will be smoke.

If the coal is too coarse, it will not pass through the upper grate when coking is completed, and the fireman will be compelled to help it through with a fire tool. If this is done at a time when uncoked coal is upon the upper grate, some of it will probably accompany the coked coal to the lower grate and there will be smoke.

If it is desired, at any time, to force the furnace to meet an extra demand for steam, the coal must either be burned directly upon the lower grate or the uncoked fuel upon the coking grate must be forced through upon the lower grate, in order to make room for a fresh supply of green coal.

To force a furnace of this type without producing smoke, is an exceedingly delicate operation. If an even load is carried, and the demands upon the boiler are not excessive, this furnace with careful firing will give good results as to smokelessness. As with most other devices, a great deal depends upon the assistance of the fireman.

137. Q.—What precautions are necessary in firing an internally-fired boiler?

A.—In firing an internally-fired boiler, the same precautions should be used as with any other type. The water level should be kept constant, the fire should be of a uniform depth and evenly distributed; also, as little cold air as possible should be allowed to enter the firebox when cleaning fires.

138. Q.—How would you handle a boiler if the safety valve stuck and the gage showed 25 lb. more than the valve was set for?

A.—Reduce pressure, and carry no higher pressure than gage is set for until the boiler can be shut down. When pressure is off and boiler cooled, remove safety valve and repair.

139. Q.—What would you do in case you found a lap leak on a boiler?

A.—In case a leak is discovered in the lap seam of a boiler, not an instant's time should be lost in determining for a certainty just what is the cause. If it is caused by a rivet, calk it when the boiler pressure has been reduced to zero or at least very low.

It sometimes happens that a boiler sheet will have a defect in it that cannot be detected but that when punched and rolled, a fracture is set up in the ligaments between two or more adjacent holes. This fracture may not assert its weakness for months or years after the boiler is placed in service, and possibly never might but for some sudden strain such as introducing cold water suddenly, or opening the fire doors to cool off, or cooling off too rapidly before washing out. At any rate, the constant expansion and contraction will always tend to enlarge such fractures until eventually the fireman may be startled to see steam or water issuing from a place where it never was before.

Examine the seat of trouble carefully and if necessary call in an experienced boiler maker. This means a sober man of years of actual experience actually making and repairing boilers where the personal factor is the all-important one. Do not take a single chance. Cut the boiler out and shut down the entire establishment and tie up a whole railroad system sooner than take a long chance and demolish the plant with another "mysterious" boiler explosion.

Give the boiler the water test, and note the nature of the leak and you can tell better the cause of it.

Sometimes removing a rivet carefully near the center of the leak will disclose the fracture in the rivet hole. It may be necessary to remove an entire course from the boiler and put in a new one, and it may be necessary only to put on a patch. At any rate, get a "boiler doctor" and don't run long chances.

140. Q.—If you discovered the water out of sight in your gage glass and you had a high steam pressure and heavy fire in the furnace, what would you do?

A.—If the water should suddenly drop out of sight and not return to its normal level, the only safe procedure would be to bank the fires at once with ashes or green coal, close the draft, cut the boiler off from the line and allow it to cool off.

141. Q.—If you had two boilers in battery with their safety valves set to blow off at 125 lb., and the inspector cut one down to 100 lb., what would you do? Both boilers are connected to a common steam main.

A.—If operating conditions are such that both boilers are run at the same time, the only thing to do under such conditions would be to carry no more than 100 lb. on both units.

142. Q.—What should be done in case of a boiler with unstayed flat surfaces?

A.—An unbraced flat surface on a boiler should not be permitted. If such a condition is found, the boiler should not be operated without having it inspected and having the strength carefully figured.

Boiler Inspection

143. Q.—Describe what you would do in making an external examination of a boiler.

A.—Pass in on the grates, noting fire sheets, circular seams and side wall linings; continue back over bridge wall and examine plates, rear head, flange and tube ends. Note blowoff pipe ell and flange on shell, and whether pipe has free opening in rear wall. Note fusible plug, also condition of brick in combustion chamber walls and at and under rear arch. Pass out setting door at rear, examine blowoff connections and valves, also lead to outside or catch basin. See if latter is properly ventilated. Pass to top of setting, and examine all steam connections and attachments. See

that safety valve is free and that column pipe and steam gage connections are properly placed. Next, open flue cap, examine front head and tube ends and head flange. Note water column and connections. Remove plugs in cross and see that pipes are clear, then test gage. In the course of the inspection, the condition of the setting walls is noted.

144. Q.—Describe what you would do in making an internal examination of a horizontal tubular boiler.

A.—With boiler open and empty; pass in through upper manhole, examine plates, flanges, seams and tubes; sound the braces; note location, thickness and consistency of scale, depth and extent of corrosion, location of oil and deposit—if any of these are present. See whether all openings to attachments are clear. Next, enter lower manhole and examine plates, tubes, circular and head seams, and head flanges; sound braces; note opening to blowoff pipe and whether nipple is screwed flush, or sufficiently, into flange.

145. Q.—In a hydrostatic test, should cold or hot water be used, and why?

A.—In a hydrostatic test, water at a temperature of not less than 125 deg. F. should be used, according to the rules of the American Boiler Manufacturers' Association. The reason for using hot water is to give the expanded conditions of the boiler, as nearly as possible.

146. Q.—What would you do on taking charge of boiler plant?

A.—Go over the entire equipment to see that everything was in proper condition for safe and economical operation. If such is not the case, make necessary repairs and changes as soon as possible.

147. Q.—What would you do when first going on watch?

A.—Blow down the boiler one gage, if there is plenty of water. Try out water column, gage glasses and low and high water alarm, if such are used; test the safety valve and see that things in general are in proper operating condition, especially on feed water supply system.

148. Q.—What should be looked for when inspecting a new boiler?

A.—Internally, the seams, tubes, heads, flanges, braces and openings to all attachments should be examined. Externally, the condition of shell plates, seams, flanges, tube ends and all outside attachments should be noted; also the setting and lining walls, lead and area of breeching, and stack opening.

149. Q.—What precautions should be taken when cleaning and inspecting a boiler?

A.—The main thing to look after before cleaning and inspecting a boiler is to see that the stop valve leading to the boiler header is closed and locked shut. The boiler should also be cold, but should not be cooled too quickly. The water must be blown out, manhole and handhole covers removed and careful inspection made before entering the boiler to make sure that there is no leakage of steam and feed valves.

150. Q.—When a man is inside a boiler, what should be done for his safety?

A.—Proper precautions before sending a man into a boiler are to be sure that there is no steam in the boiler, or possibility of hot gases getting into the boiler. Then lock the main steam, feed and blowoff valves to the boiler and give the man the keys to the locks.

151. Q.—How would you clean and inspect a boiler that was set in a battery with other boilers?

A.—To clean and inspect a boiler of a battery, close off the steam and feed valves, lock these valves closed. and go ahead as if the boiler were separate. The boiler should be cooled down, the water blown out, manhole and handhole covers removed, and careful inspection made before entering the boiler, to make sure that there is no leakage of steam or feed valves.

152. Q.—What defects would you look for when cleaning a boiler?

A.—Scale, bags, corrosion and cracks.

153. Q.—How would you determine the point of leakage from an internal view of boiler, and how would you determine if inside plate was cracked or rivet broken from viewing the external part of the boiler where leakage was?

A.—From collection of scale or deposit at the point, in the water space, from discoloration of material about same, in steam space. By leakage, by sound, or in case of rivet by looseness.

154. Q.—How can a broken stay-bolt be located?

A.—If the stay bolt is hollow it will leak if broken, but if it is solid, the best way to find out is to go inside and by tapping lightly with a hammer, you can tell by the sound whether it is broken or not.

155. Q.—How can the true thickness and strength of the plate be found in case of a pitted boiler?

A.—If the pitting covers considerable area, a hole can be drilled in the sheet through which a micrometer or other convenient device can be inserted and the sheet measured; after which the hole can be tapped and plugged, or a rivet inserted. A depth gage is a convenient and handy device to measure depressions in a boiler sheet also and is easily made. Having found the thickness of the sheet, the same working pressure must be calculated from the thinnest place found, for the reason that a chain is no stronger than its weakest link.

156. Q.—What are the common causes of boiler explosions?

A.—(a) Sheets that have become weakened through corrosion and pitting. (b) Sheets that have been weakened by burning; brought about by low water; or by allowing a heavy coating of scale to form on the inside thus preventing the water from coming directly in contact with the sheet, while at the same time the outside of the sheet was subjected to intense heat. (c) Subjecting the boiler to a higher pressure than it was designed for. This may be due either to the safety valve refusing to open at the proper time on account of its being stuck, or the safety valve may have been intentionally set for too high a pressure. (d) Insufficient bracing.

157. Q.—How would you make an honest and full monthly report of the conditions of your boiler?

A.—If this report is to be made on all the boilers every month, you should have some form of blank of suitable size for filing, to be filled in every month. The items on this blank would depend on the circumstances and would include the following: Condition of furnace,

side walls and grates or stoker; repairs needed, and repairs made since last report; condition of tubes and drum, repairs made since last report and repairs needed; whether the surfaces were clean, dirty, scaled, bagged, etc.; the number of hours the boiler was in service and out of service, time cleaned, time consumed in cleaning and by whom cleaned; supplies used, such as tubes, gaskets, water glasses, valves, etc.; general condition as compared with the last report, also mention of any trouble or unusual condition that has not already been treated or that has occurred since last report. If some part of the boiler was in bad condition, such as bagged tubes, cracked setting, oil or pitting in drum, make especial mention of this condition and its action since last report. If a CO_2 apparatus is used, test each boiler separately at frequent intervals and note the result on the monthly report; also the coal used, ashes removed and water evaporated; also readings of the steam flow meter, draft gage, etc., if these are used. If a mechanical stoker is used, report its condition, repairs needed and repairs made. The form should also contain any other information that the individual case might make desirable. The size of report and information noted on this report depends entirely on the local conditions and object in view.

Boiler Fuels

158. Q.—What horsepower can be obtained from 1 lb. of coal having a heat value of 14,000 B.t.u.?

A.—We will first consider the theoretical side of this problem, then take up the practical side, for the purpose of comparison.

One B.t.u. is equal to 778 ft. lb. of work, so that 14,000 B.t.u. are equal to 10,892,000 ft. lb. As one horsepower is equivalent to 33,000 ft. lb. a minute, if burned one minute's time, 10,892,000 ÷ 33,000 or 330 hp. will be furnished. A pound of coal burned in 1 hr. would give 1/60 of this, or 5.5 hp.

As stated above, this is the theoretical solution of the problem, and does not take into consideration any losses whatsoever existing between the coal pile and the engine, which is an impossible condition.

We ordinarily consider an evaporation of 8 lb. of water per pound of coal very good, and under average conditions allow the following water rate for the various

types of engines, cutting off at ¼ stroke: Simple noncondensing, 30 lb.; compound condensing, 17 lb.; triple expansion condensing, 14 lb. For every pound of coal burned, per hour, a simple noncondensing engine would develop 8/30 of a horsepower an hour; a compound condensing engine, 8/17, and a triple expansion engine, 8/14 of a horsepower.

159. Q.—What are the relative values of coal and oil as fuels?

A.—These values are as shown in the table following:

RELATIVE VALUES OF COAL AND OIL FUEL

Gross Boiler Efficiency with oil fuel	Net boiler Efficiency with oil fuel.*	Net evaporation lb. from and at 212 deg. F. per lb. of oil.	Water Evaporated From and at 212 Deg. Fahr. Per Pound of Coal							
			5	6	7	8	9	10	11	12
			Pounds of Oil Equal to 1 Lb. of Coal.							
73	71	13.54	0.3693	0.4431	0.5170	0.5909	0.6647	0.7386	0.8124	0.8863
74	72	13.73	0.3642	0.4370	0.5099	0.5827	0.6556	0.7283	0.8011	0.8740
75	73	13.92	0.3592	0.4310	0.5029	0.5747	0.6466	0.7184	0.7903	0.8621
76	74	14.11	0.3544	0.4253	0.4961	0.5670	0.6378	0.7087	0.7796	0.8505
77	75	14.30	0.3497	0.4196	0.4895	0.5594	0.6294	0.6993	0.7692	0.8392
78	76	14.49	0.3451	0.4141	0.4831	0.5521	0.6211	0.6901	0.7591	0.8281
79	77	14.68	0.3406	0.4087	0.4768	0.5450	0.6131	0.6812	0.7493	0.8174
80	78	14.87	0.3363	0.4035	0.4708	0.5380	0.6053	0.6725	0.7398	0.8070
81	79	15.06	0.3320	0.3984	0.4648	0.5312	0.5976	0.6640	0.7304	0.7968
82	80	15.25	0.3279	0.3934	0.4590	0.5246	0.5902	0.6557	0.7213	0.7869
83	81	15.44	0.3238	0.3886	0.4534	0.5181	0.5829	0.6447	0.7125	0.7772

* Net efficiency = gross efficiency less 2 per cent for steam used in atomizing oil.

Heat value of oil = 18,500 B.t.u.

160. Q.—Could you get more steam out of boilers fired with soft coal by mixing crude oil with the coal?

A.—Soft coal has a heat value of about 12,000 to 14,000 B.t.u. per lb., depending on the locality from which it comes, and crude oil a heat value of about 20,000 B.t.u. per lb.

If you could get the same efficiency of burning, adding crude oil to the soft coal would give you a greater steam output; but you would have a great deal of difficulty in burning, for the conditions suited for burning soft coal economically are not those for burning crude oil.

One of the difficulties in handling soft coal is to get complete combustion of the volatile gases, and if crude oil were added, it would simply increase the volume of volatile gases, therefore increasing the difficulty of burning. The proposition is not one which can be recommended.

161. Q.—What are three important conditions that must exist in a steam boiler furnace to obtain complete combustion?

A.—There must be sufficient grate surface so that the required amount of coal can be burned without forcing.

The travel of the combustible gases must be sufficiently long so that combustion can be complete before the gases come in contact with the metal surfaces of the boiler.

A proper amount of air is necessary to produce complete combustion.

162. Q.—How much air is required to furnish the oxygen required to burn completely 1 lb. of carbon?

A.—11.52 lb. of air are required to furnish the oxygen chemically required for the complete combustion of one pound of carbon.

163. Q.—If with ordinary draft, double the theoretical amount of air is used to burn a fuel, how many cubic feet of air would be needed to burn 115 lb. of coal having a composition 5 parts hydrogen, 90 parts carbon, 3 parts oxygen, 2 parts ash?

A.—Hydrogen burns to water (H_2O), and carbon to carbon dioxide (CO_2). The ashes do not burn, so do not require oxygen, and the oxygen which is in the coal goes to combine with either the hydrogen or the carbon.

The atomic weight of hydrogen is 1, of oxygen 16. The molecular weight, therefore, of water is 2×1 or 2, plus, 16, giving 18, that is, there is 8 times as much oxygen, by weight, in water as hydrogen.

The atomic weight of carbon is 12. The molecular weight, therefore, of carbon dioxide is $12 + (2 \times 16)$ or 44, or there is $2\frac{2}{3}$ times as much oxygen in carbon dioxide as carbon.

In one pound of coal there is 0.05 lb. of hydrogen, according to the analysis which you gave. Multiplying this by 8 gives 0.4 lb. of oxygen which is required to burn the hydrogen.

In one pound of coal there is 0.9 lb. of carbon, according to analysis, and multiplying this by $2\frac{2}{3}$ equals 2.4 lb. of oxygen required to burn the carbon. Adding 0.4 and 2.4 gives 2.8 lb. of oxygen required for each

pound of coal. The coal itself furnishes 0.03 lb., therefore, the air must furnish 2.8 — 0.03 = 2.77 lb. of oxygen.

You state in your problem that twice the amount of air theoretically required is to be supplied. Multiplying, therefore, 2.77 by 2 gives 5.54 as the pounds of oxygen which must be supplied by the air. Since free air contains 23 per cent oxygen, the total weight of the air required for each pound is (5.54 ÷ 23) × 100, giving 24.5 lb. as the weight of the air required to burn one pound of coal under the conditions specified. To burn 115 lb., will take 115 × 24.5 or 2,770 lb. of air. Since one pound of air occupies a space of 13.3 cu. ft., nearly, the volume of air required to burn the 115 lb. of coal is 2,770 multiplied by 13.3 or 36,900 cu. ft.

164. Q.—How is the amount of air to burn 1 lb. of fuel determined?

A.—Letting A represent the weight of air required per pound of fuel, C, H and O the proportional part by weight of carbon, hydrogen and oxygen in the fuel, the value of A may be readily determined by means of the following formula:

$$A = 34.56\left(\frac{C}{3} + H - \frac{O}{H}\right)$$

When the flue gas analysis is known, the air actually supplied may be approximately determined from this equation:

$$A = 3.032\left(\frac{N}{CO_2 + CO}\right) \times C$$

Where A is as above; N. CO and CO_2 the percentages by volume of nitrogen, carbon monoxide, and carbon dioxide in the flue gas, and C is the proportional part by weight of carbon in the fuel.

165. Q.—What is the effect of admitting too little air to the boiler furnace?

A.—Insufficient air in the furnace means that combustion will not be completed, hence the gases will contain a large per cent of carbon monoxide and a small per cent of oxygen. Incomplete combustion and a loss of heat will result.

166. Q.—If complete combustion of coal is not secured, what proportion of its heat is lost?

A.—The heat lost due to the fact that the carbon in the fuel is not completely burned, and carbon monoxide is present in the flue gases, in B.t.u. per pound of fuel is:

$$\text{Loss} = 10{,}190 \times \left(\frac{\text{CO}}{\text{CO} + \text{CO}_2} \right)$$

where CO and CO_2 are percentages by volume of the carbon monoxide and carbon dioxide in the flue gases. One pound of carbon burned to CO_2 yields 14,540 B.t.u.; if burned to CO, it yields only 4350 B.t.u., a loss of 10,190 B.t.u.

167. Q.—How many heat units are required to raise the temperature of 1 lb. of air from 70 to 2000 deg. F.?

A.—In raising the temperature of 1 lb. of air from 70 to 2000 deg. F., the required temperature rise is equal to 2000—70 or 1930 deg. F. Air, at constant pressure, has a specific heat of 0.2375, so that the total heat required would be equal to 0.2375 times 1930, or 458.38 B.t.u.

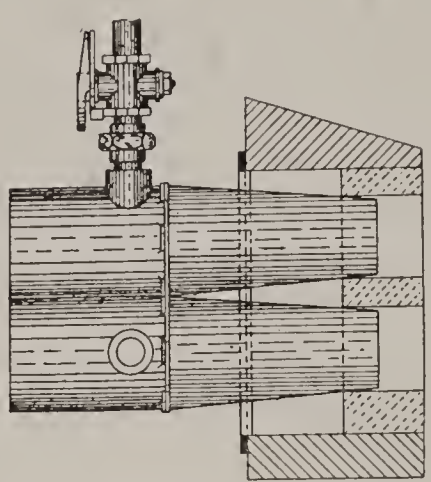
168. Q.—What is a fair average consumption of natural gas per horsepower-hour for firing a steam boiler?

A.—J. M. Whitham reports in the transactions of the A. S. M. E., the results of several tests of water-tube boilers with natural gas. With a Cook vertical boiler, 44.9 and 41.0 cu. ft. of gas per boiler horsepower-hour were consumed; with a Heine, 46, 40.7 and 38.3 cu. ft. per boiler horsepower-hour; with a Cahall vertical, 42.3 and 34 cu. ft.

169. Q.—How should natural-gas burners be installed under a 66-in. by 16-ft. boiler now arranged for hand firing with coal? How large should the combustion space be and what kind of baffling should be used?

A.—Natural gas is best burned by the use of a large number of small burners, each being capable of handling approximately 30 rated boiler horsepower. This obviates the danger of any laning or blow-pipe action of the gases which might be present if large burners were used.

Where natural gas is the only fuel, the burners should be evenly distributed over the lower portion of the boiler front. If, however, this fuel is used as an auxiliary to coal, the burners may be placed through the fire doors.



A169. INSTALLATION FOR NATURAL GAS

A large combustion space gives the best results, say 2 cu. ft. per rated horsepower. A checker-work wall is sometimes placed in the furnace about 3 ft. from the burners to break up the flame, but with a good type of burner and furnace this wall is not necessary. Good results are secured with the burners inclined slightly downward toward the rear of the furnace where the gas is burned alone and there are no coal-fired grates.

170. Q.—Will you please give some information relative to the use of natural gas under steam boilers?

A.—Information available on burning of natural gas under boilers is not very complete, as this method of generating steam is, as you know, a proposition which applies to only a comparatively small part of the country, and even there is likely to be expensive.

The best practice, so far as recorded, is, first, to have a burner which gives a rotary jet, in order to mix thoroughly the gas and air as it leaves the burner. The burner should be set so as to avoid the direct flame striking the tubes. This is usually accomplished by putting a brick arch above the burner, extending out some 18 in. to 2 ft. beyond the end of the burner. The burner is set so that the flame goes towards an enlarging combustion space.

If the boiler is like the Stirling, the jet is set at the front end and arranged to discharge towards the boiler

—a checker-work bridge wall being installed at some 4 ft. from the jet. The grate is covered with fire-brick, the back half being laid close, and the brick under the jet being laid with air spaces running in a fan-shape, from 7/8 in. directly under the burner to about 3/8 in. at the end of the fan-shaped space. Outside this fan-shaped section, the furnace on the front half of the grate has an air space of 1/8 in. between bricks and a space of 5/8 in. is left all around the side walls.

For a boiler like the Babcock & Wilcox, the burner is better set at the back end of the furnace, discharging towards the front, with a checker-work at about 4 ft. in front of the jet, and baffle brick extending on the under sides of the tubes to some 2 ft. in front of the jet. A combustion space should be provided, 3/4 of a cubic foot per rated horsepower.

Tests on boilers fired with natural gas have shown that at 140 lb. pressure, with feed water 116 deg. F., a gas pressure of 8 oz., and a draft in the base of the stack of 0.35 in. of water, the flue gas temperature shows 417 deg.; the evaporation, actual, is 0.78 lb. of water per cubic foot of gas, and the equivalent from and at 212 deg. F. is 0.895 lb. of water per cubic foot of gas.

Feed Water

171. Q.—With feed water containing considerable scale-forming material, what can be done to the boilers themselves to keep them in good condition?

A.—The best way to handle any boilers where the feed water is bad, is to treat the water before it enters the boilers, pass it through a feed-water heater and purifier, then see that the water is fed in so that it will first pass to the mud drum or to the rear of the boiler, where the sediment will be deposited and can be blown out. The point is to get the scale forming material deposited in a soft mud instead of a hard scale, and to arrange the circulation so that this mud will be thrown down as far as possible from the hottest part of the boiler, so that it will not bake onto the plates.

172. Q.—If foaming is caused by dirty feed water, just what is the action or condition of the water in the boiler?

A.—The action is more mechanical in nature than chemical. This same action is found in the case of an

ordinary coffee pot. If you take a pot of coffee still containing the grounds and set this upon a flame, you will notice that the coffee in the pot will, when it becomes sufficiently hot, rise to a greater height than would a like amount of water under the same heat. This will explain to you how dirty feed water will cause a boiler to foam.

173. Q.—How would you handle a boiler that was foaming badly?

A.—Do not allow water to rise above 2 gages; blow down frequently and do not force more than absolutely necessary.

174. Q.—How can foaming be remedied without closing down boiler or engine?

A.—In case of foaming, close throttle, and keep closed long enough to show true level of water. If that level is sufficiently high, feeding and blowing down will usually suffice to correct the evil.

In case of violent foaming, caused by dirty water, or change from salt to fresh, or vice versa, in addition to the action above stated, check draft and cover fires with fresh coal.

175. Q.—Our boilers prime badly the latter part of each week. Closing the throttle valve to the Corliss engine partly, somewhat reduces the trouble. Why is this so?

A.—As to cutting down the water going to your cylinder by partly closing the throttle, this is a case of wire-drawing. By closing the valve, you reduce the pressure in your cylinder. In this passing of steam through a small opening, permitting the pressure to drop reduces the heat in each pound of saturated steam. The heat thus given up by the steam is used in evaporating the moisture carried by the steam, thus reducing the amount of moisture.

176. Q.—If you had a boiler badly scaled, how much soda ash would you use, that is, is there any rule in regard to so much per horsepower? Or do you have to experiment to find the right amount?

A.—We presume that you mean to prevent the formation of more scale in your boilers. It is found that a small quantity of soda ash will act with good

results on large quantities of feed. If too much soda ash is used, it is liable to promote priming, with all the dangers as well as the inconveniences which accompany it.

The best method is to connect the feed pump or injector to a soda tank, so that a supply of soda solution can be drawn at regular intervals. The proper amount is usually determined by experience, but ordinarily varies between 1 and 2 lb. per day for the average boiler. The soda will not injure the boiler unless it is impure and contains acids.

177. Q.—When boiling soda ash in a boiler to remove any oil which may be present, would you leave the manhole plate out on top or not, and why?

A.—In boiling soda ash in boilers, it is usually allowed to act with a pressure not to exceed 15 lb., and thus it is necessary to keep the manhole closed.

178. Q.—What is the objection to the use of sal-ammoniac for the precipitation of carbonate of lime in feed water.

A.—Sal-ammoniac (ammonium chloride NH_4Cl) is made by passing ammonia gas through hydrochloric acid (HCl) and when dissolved in water and boiled a portion of the ammonia escapes as steam leaving an acid solution which would be dangerous in a boiler.

179. Q.—What is the result of poor circulation of water in a boiler?

A.—Poor circulation results in the sticking of steam bubbles to heating surfaces, which will mean poor efficiency in the boiler. It may mean, also, in case of a water-tube boiler, that the steam bubbles will form in the tubes, and when they break at the surface, will throw a large amount of water into the steam space, thus causing very wet steam or what is known as "priming."

180a. Q.—What is the best remedy to stop internal rusting of boilers that are not in continual operation, but left empty several months at a time?

A.—If a boiler is to remain idle for some time its deterioration may be much more rapid than when in service. If the period for which it is to be laid off is not to exceed three months, it may be filled with

water while out of service. The boiler should first be cleaned thoroughly, internally and externally, all soot and ashes being removed from the exterior of the pressure parts and any accumulation of scale removed from the surfaces. It should then be filled with water, to which five or six pails of soda ash have been added, a slow fire started to drive the air from the boiler, the fire drawn and the boiler pumped full. In this condition it may be kept for some time without bad effects.

If the boiler is to be out of service for more than three months it should be emptied, drained and thoroughly dried after being cleaned. A tray of quicklime should be placed in each drum, the boiler closed, the grates covered and a quantity of quicklime placed on top of the covering. Special care should be taken to prevent air, steam or water leaks into the boiler or on to the pressure parts, to obviate danger of corrosion.

180b. Q.—Can a mixture of coal oil and graphite be used to remove or prevent scale in a horizontal tubular boiler?

A.—The introduction of crude oil or kerosene into a boiler has, from time to time, been used as a means of preventing scale formation on the heating surfaces, but this use of kerosene or crude oil is not to be recommended. While cases may arise in which boiler water can be effectively treated within the boiler itself, oil is not the reagent to be used. The distilling off of the lighter oils finally leaves a heavy, gum-like carbonaceous deposit on the heating surfaces, which will tend to cause a burning of the affected parts. Further, such oils may contain minerals which will saponify where the feed is sufficiently alkaline, and severe foaming will result.

Graphite alone is, however, used extensively for the removal of scale in steam boilers, the action being of a mechanical nature rather than chemical. Only very finely ground graphite of good quality should be employed.

No set rules can be given for the amount to use because this will depend upon local conditions, such as the amount and nature of the scale-forming material in the water. Each operator should study his own requirements, but experience has proved the following schedule quite satisfactory:

For each boiler each day of 12 hr.:

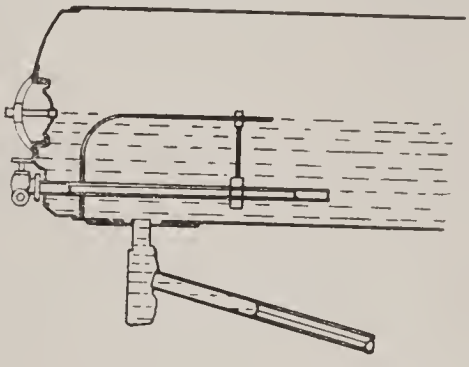
Up to 100 hp., use 2/5 pint.
Up to 150 hp., use 3/5 pint.
Up to 200 hp., use 4/5 pint.
Up to 250 hp., use 1 pint.
Up to 300 hp., use 1 1/5 pints.
Up to 350 hp., use 1 2/5 pints.
Up to 400 hp., use 1 3/5 pints.
Up to 450 hp., use 1 4/5 pints.
Up to 500 hp., use 2 pints.

After all the old scale has been removed, the daily injection may be decreased slightly. It is the regular introduction of small amounts of graphite that brings about the most satisfactory results. In addition to the above, put about 2 qt. of graphite into the boiler after each cleaning.

Feed Water Piping

181. Q.—Where does the feed pipe enter a B. & W. boiler and how is it supported?

A.—The feed pipe in a B. & W. boiler enters the boiler through the lower part of the front drum head,



A181. Feed pipe connection to B. & W. boiler

and is extended beyond the front tube header through the baffle plate at the head of the boiler and is, therefore, partly supported by the latter. It is also supported by a small stay bolt near the end, which is attached to the feed pipe by a coupling and extended upwards through the baffle plate, to which it is fastened by a nut.

182. Q.—I have a 30 hp. boiler with the feed water pipe entering the front end, just above the top row of tubes. Can I enter the water through the blowoff pipe, as a red sediment forms on the tubes, which I want to get rid of?

A.—There is considerable difference of opinion as to the advisability of feeding the boiler through the blowoff. It is done in many plants, and if there is not much formation of scale, it probably is not harmful. If there is formation of scale, it tends to produce a current which will carry the sediment away from the blowoff and toward the front end of the boiler, where it will lodge over the fire, and will be more likely to produce trouble than if carried toward the back end of the boiler. Also, putting the water in through the blowoff gives no chance for heating the water inside the boiler before it is discharged against the tubes.

A plan which will be better than this would be to carry your feed pipe from its present ending inside the boiler, back well to the rear end, and then turn it down, either between the rows of tubes in the center, or carry it across to one side and then down along the shell of the boiler, discharging near the bottom.

In this way, by its passage through the feed pipe inside the boiler, the water becomes heated practically to boiler temperature before it is discharged to mingle with the circulation of the boiler. You can bring the discharge to any point where you prefer to have the sediment deposit, but naturally this would be near the blowoff.

183. Q.—With two or more boilers in battery, could one be fed from another through the blowoff?

A.—With the two boilers under equal working pressures and set at the same level, only enough water would pass from one boiler to the other to cause the water in both boilers to stand at the same level. It would be undesirable to feed in this way, but might be done for an emergency.

184. Q.—What changes would you make to stop the vibration and pounding of the feed line to the boiler?

A.—Vibration and pounding in the feed line is caused by the inertia of the water which tends to continue in motion after the impelling force has ceased to act, and its energy of motion is expended in giving a blow to the containing pipe. Any means which will induce an uninterrupted flow or will bring the water gradually to rest will prevent the water hammer. In a

duplex pump, the first result can be obtained by adjusting the steam valves so that one piston begins its stroke before the other one has stopped.

The second result (gradually checking the flow) can be secured by placing an air chamber on the discharge pipe near the end of the pump or by regulating the pump cushion in such a manner as to bring the piston gradually to rest rather than holding its speed to the end of the stroke, then stopping abruptly.

Oftentimes the noise accompanying water hammer in the feed line can be eliminated by making the pipe more secure at its points of support.

185. Q.—How should the stop valves on a feed water line be placed?

A.—The valves on the feed line to a boiler should be placed so that they close against the flow of the water to the boiler. When the flow ceases, the valve tends to close from the boiler pressure, and in case the disc comes loose from the stem, the flow will hold it open so long as the feed is operating and the pressure will close it when the feed stops.

186. Q.—If the disc of the feed line check valve was lost could the boiler still be operated?

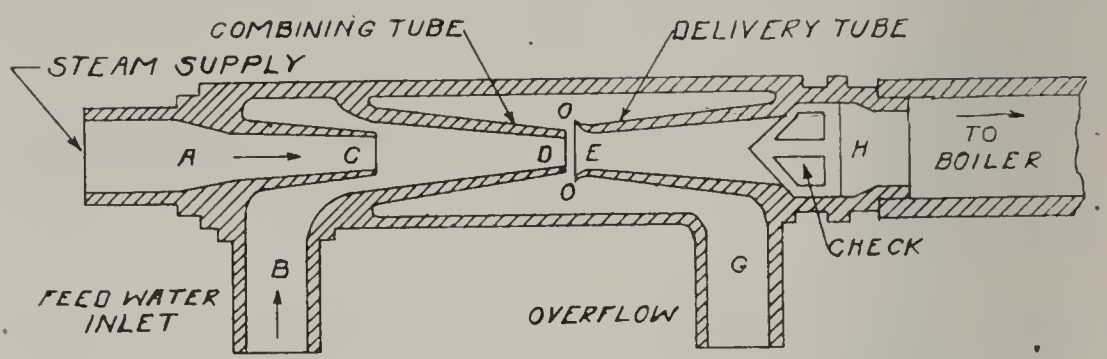
A.—If the disc of the check valve comes loose, the boiler can still be operated, provided the pump valves are tight, since the discharge valves of the pump will prevent backing up of water from the boiler. Whenever the check valve is inoperative, however, if it becomes necessary to shut down the pump or injector, the stop valve in the feed line should be closed, and care should be taken to see that this is opened again when the pump or injector is started.

187. Q.—What should be the height of a standpipe to feed a boiler carrying 60 lb. pressure?

A.—Each foot of head in a column of water produces a pressure of 0.43 lb. per square inch. The number of feet to equal 60 lb. pressure would then be $60 \div 0.43$, which gives 139.5 ft. In addition to this, a stand pipe to feed a boiler would have to have sufficient pressure to overcome the friction in the pipes and to give the feed water velocity as it enters the boiler.

188. Q.—Upon what principle does the injector operate?

A.—Figure A188 is a diagram of the simplest form of a single tube injector. Boiler steam admitted at A and, flowing through nozzle C and combining tube D to the atmosphere through G, partially exhausts the air



A188. Section of single tube injector

from pipe B, thereby causing the water to rise until it comes in contact with the steam. The steam emerging from nozzle C at high velocity condenses on meeting the water and imparts considerable velocity to it. The energy in the rapidly moving mass is sufficient to carry it across opening O, lift check H from its seat and force it into the boiler. The steam then ceases to escape at G. Usually a check valve is provided in G which closes when the velocity of the water current causes a partial vacuum in the chamber around tubes D and E.

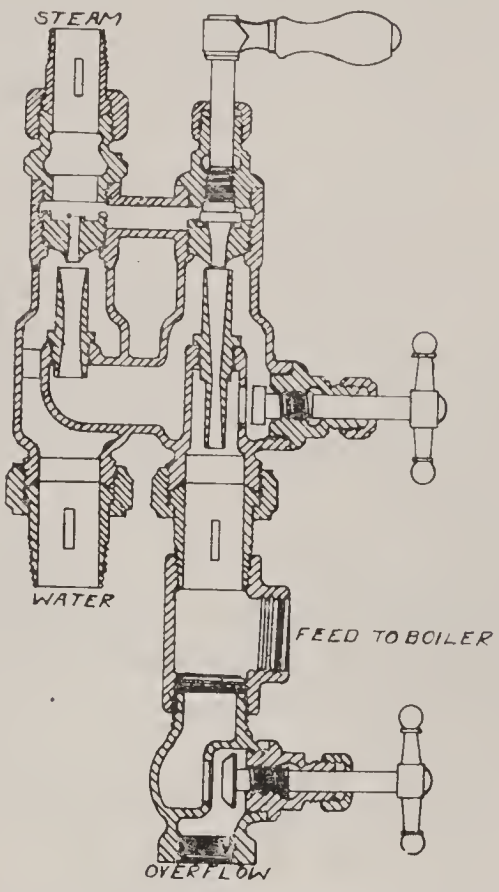
189. Q.—What is a double tube injector? What advantage has it over a single tube injector?

A.—The double injector has two steam tubes, one tube for drawing up water, and a second tube for forcing water into the boiler. Its advantage over a single injector is that it can handle hotter water and higher lifts.

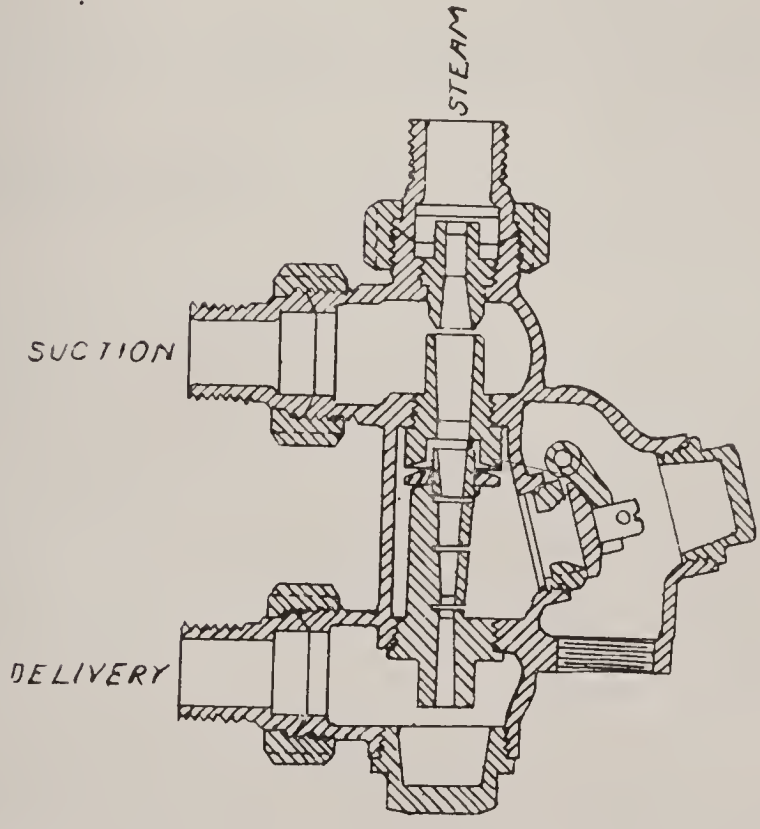
190. Q.—What is an automatic injector?

A.—The automatic injector has a movable part, which, as the steam jet gets into action and creates a suction between the forcing tube and the lifting tube, is drawn up and closes the outlet to the overflow. Until this suction is formed by the action of the steam and water jet, the movable piece will rest, so that water will escape from the overflow. In case the stream into the boiler breaks for any reason, the movable piece will drop back and water will again escape from the over-

flow until the jet is established. The advantage is, that the injector, after breaking, will restart of its own accord instead of having to be shut off and restarted by hand.



A189. SECTION OF DOUBLE TUBE INJECTOR



A190. SECTION OF AUTOMATIC INJECTOR

191. Q.—How hot can water be delivered into the boiler by a double tube injector?

A.—Under ordinary operating conditions, a double tube injector will deliver water to the boiler at from 160 to 180 deg., but under some special laboratory tests, conducted with the idea of learning what could be done, temperatures a number of degrees higher than boiling point were reached.

192. Q.—Can an injector be used to advantage with an open or a closed heater?

A.—The water from an injector may be passed through a closed feed water heater before entering the boiler. An open feed water heater cannot be used in conjunction with an injctor if the feed water is heated to a practical degreee.

193. Q.—Why should an injector be used with capacity larger than required for regular feeding?

A.—An injector has no overload capacity. The speed or action cannot be varied. It must either run full blast or not at all. Hence it must have a good deal more capacity than a boiler feed pump, which can be speeded up to take care of overloads.

194. Q.—What is a simple method of cleaning the tubes in an injector?

A.—Immerse the injector for 24 hr. in a solution of 1 part muriatic acid to 6 parts water.

195. Q.—Could you feed a boiler under 150 lb. pressure with an inspirator that was taking steam from a boiler under 100 lb. pressure?

A.—It would be possible to feed a boiler of 150 lb. pressure from a boiler carrying 100 lb. of steam. Commercial inspirators are usually arranged to feed against 40 to 50 per cent above the pressure actuating them. Ordinarily this takes care of the wear, friction of valves, etc., so that in the above case, where extra over-pressure is required, it would come under the head of a special injector, which would be made by changing the forcer steam nozzle.

Boiler Room Accessory Apparatus

196. Q.—What is the rule for figuring the proper size of safety valve to be used on a boiler?

A.—The Boiler Code of the American Society of Mechanical Engineers gives a table for the steam discharging capacity of safety valves at various pressures, based on the heat generated from fuel burned, a boiler efficiency of 75 per cent and Napier's formula for flow of steam through an orifice into atmosphere.

For valves with 45 deg. bevel seat the formula is:

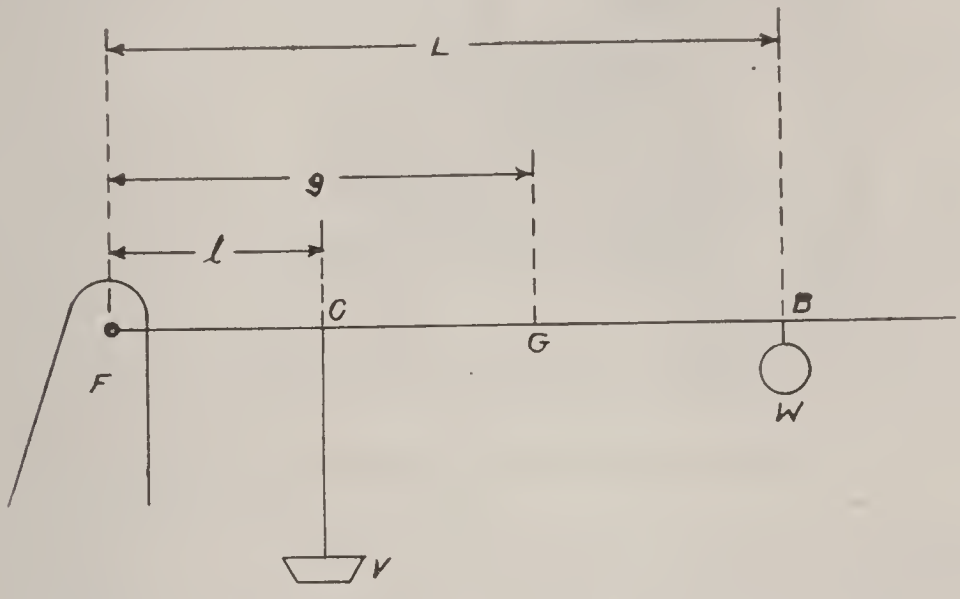
$$CH = 160{,}856\ PDL$$

For valves with flat seat at 90 deg.:

$$CH = 227{,}487\ PDL$$

where C = total fuel burned per hr. (lbs. or cu. ft.)
H = heat of combustion (B.t.u. per lb. or cu. ft.)
D = diam. of valve seat, in.
L = vertical lift of valve disc, in. measured after the sudden lift due to pop.
P = absolute boiler pressure, lb. (gage + 14.7).

197. Q.—What is the method for figuring the pressure at which a lever safety valve will open? How are the required weight of the ball and length of the lever determined?



A197. ARRANGEMET OF LEVER SAFETY VALVE

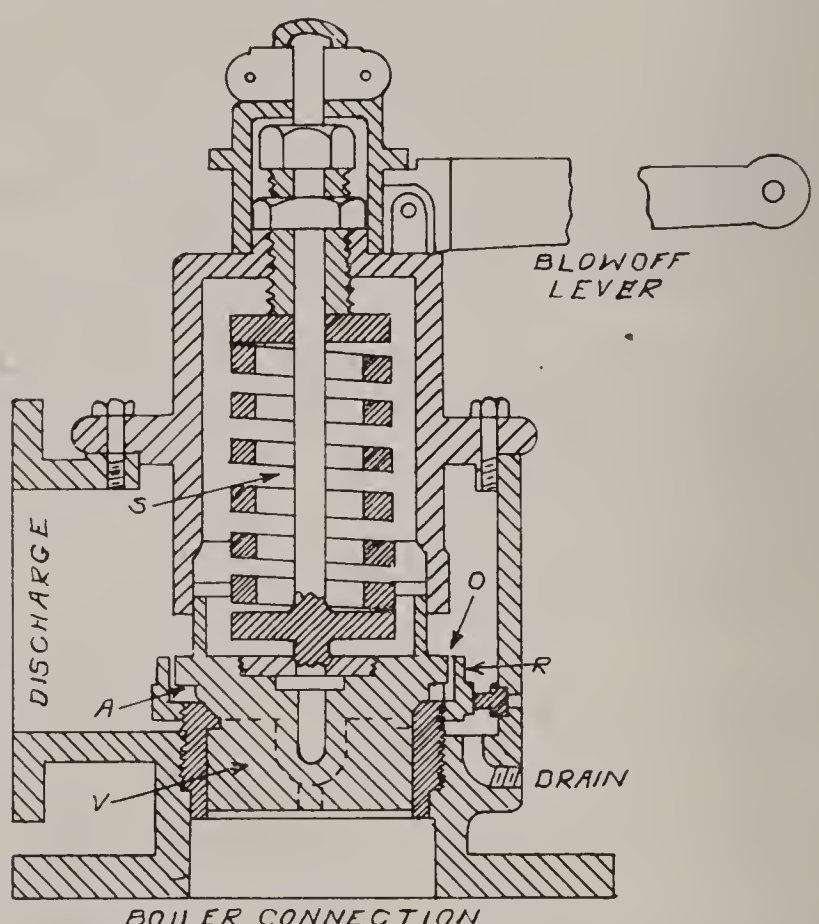
A.—Figure A197 indicates the arrangement of this type of safety valve.

The valve stem and weight are attached to the lever at C and B, respectively; the fulcrum is at F.

Let L = distance from fulcrum to ball, in inches;
g = distance from fulcrum to center of gravity of lever, G, in inches;
l = distance from fulcrum to center of valve stem in inches;

A = area of orifice beneath bottom of valve in inches;
W = weight of ball, W, in pounds;
V = weight of valve and stem, in pounds;
w = weight of lever, in pounds;
P = blowoff pressure, in pounds per square inch.

It is a necessary condition of equilibrium that the algebraic sum of the moments of all forces about a given point shall equal zero. Hence, taking F as the center of moments and treating all the forces acting downwards



A198. SECTION OF POP SAFETY VALVE

as positive and the upward pressure of the steam as negative, we have,

$$PA \times l = (W \times L) + (w \times g) + (V \times l)$$

Solving this equation for P gives,

$$P = (WL + wg + Vl) \div Al$$

Solving for W gives,

$$W = (PAl - wg - Vl) \div L$$

Solving for L gives,

$$L = (PAL - wg - Vl) \div W$$

P is, of course, the pressure at which the boiler is to be operated. A is determined according to the rules of the A. S. M. E. code for the volume of steam to be discharged; and the other values may be varied to suit

the judgment of the designer. For a valve already made, the length, L, at which the ball is placed will vary according to the pressure to be carried.

198. Q.—Why does a pop safety valve pop?

A.—Figure A198 is a cross section of a pop safety valve. The boiler pressure acting upon the underside of valve V is resisted by a compression spring S. As soon as the boiler pressure exceeds the pressure of the spring, the valve lifts from its seat and the steam escapes through the opening O. The static pressure of the steam plus the force of its reaction in being deflected from the surface A, holds the valve open until the pressure of the boiler drops about 5 lb. below that at which the valve is lifted. The additional area of valve exposed to pressure when the valve lifts, causes it to open with a sudden motion, which has given it its name, and it also closes suddenly when the pressure has fallen. These valves are so arranged that the spring tension may be varied without taking them apart. A provision is made for lifting the valve by means of a blowoff lever.

199. Q.—How does the exhaust heat the water in an open heater?

A.—In an open heater the water is heated by direct contact with the exhaust steam, the two mingling in the open space of the heater. Usually the water passes downward in a spray or overflows a series of pans, the steam entering near the bottom and passing upward.

200. Q.—Can an open heater be used with an exhaust steam heating system?

A.—You can use an open heater with any kind of a heating system. The writer has used one in connection with a vacuum heating system, on which a vacuum of 10 in. was maintained constantly, with all the way from 1 lb. vacuum to 3 lb. back pressure in the main supply close to the heater. Live steam was used when no exhaust was available, and in case of shut down of the vacuum pump for any reason, straight back pressure was used for the time being.

201. Q.—What prevents the water from backing up the exhaust pipe into the engine when an open heater is used?

A.—The overflow valve in the heater will prevent water in the heater from rising to the level of the exhaust pipe flange. There is or should be a drip valve in the lowest point of the exhaust pipe which should be open while starting the engine.

202. Q.—What temperature can be reasonably expected of discharge from an open heater with proper size piping at 1 lb. pressure?

A.—This depends upon the amount of steam available, and the amount of water which it is required to heat. If, however, these values are properly proportioned, or the heater is being worked at its rated capacity, it is possible to obtain a temperature of about 210 or 212 deg. F.

203. Q.—In our open heater, of pan type, with water level regulator, the regulator sometimes does not work and the heater fills with water which blows out of the exhaust head with the steam. There is a gage on the exhaust pipe between engine and heater, which when the heater overflows shows 15 lb. pressure. Is there danger of blowing out a cylinder head or otherwise injuring the engine?

A.—The trouble seems to be in the water regulator which does not close as it should when the water has reached a certain height. On a well designed open heater, that is, one designed with a view of safety and continuity of operation, means should be provided for letting the excess water out of the heater before it can run over into the exhaust pipe.

There is always a chance for a chip or some other substance passing in with the water and lodging under the regulating valve, thus preventing its closing entirely. The pressure of 15 lb. on the heater (and consequently on the engine, as back pressure), is caused by the water passing up through the exhaust pipe in intermittent slugs of steam and water, in a manner similar to the discharge of water from an air lift well.

As far as the danger to the engine is concerned, there is no actual danger as long as the engine is operating under a full and steady load, for the passage of the exhaust steam will always tend to keep the engine free from water. If the load is light or the steam pressure excessively high, the steam is likely to expand down

below the atmospheric line, when, as soon as the exhaust valve opens there is a danger of water from the heater being drawn up into the cylinder with disastrous results.

By all means remove this danger at the earliest possible moment, as a backpressure of 15 lb. on an engine is not conducive to economical operation, to say nothing of the danger should the load fall off at a time when the heater is full and overflowing.

204. Q.—In what types of closed heaters does the steam pass through the pipes or coils instead of around them?

A.—In the Otis, Baragwanath, Berryman, Goubert and Murray types of feed water heaters, the steam is passed through the tubes instead of surrounding the tubes, as in the other type.

205. Q.—How much above 212 deg. can the feed water be heated in a closed heater?

A.—Where the feed water heater is connected in the line between the feed pump and the boiler, it is possible to get a temperature of 300 deg. F.

206. Q.—What are two good points of a closed heater?

A.—With a closed heater it is not possible for any oil in the exhaust steam to get into the boiler as it does not mingle with the exhaust steam. With this type of heater it is possible for the pump to take water from any reasonable source even to 20 ft. or more below the level of the suction inlet, and force it through the heater into the boiler.

207. Q.—How can leaks in a closed heater be detected?

A.—Probably the most apparent indication would be an increased amount of condensation. This, however, might be so gradual that you would not notice it for some time. A good plan is to test the heater occasionally by raising the water pressure in it and keeping the steam out, any water then coming from the drips will come through leaks in the heater.

208. Q.—Why should a feed pump discharge through a closed heater rather than drawing water from the heater?

A.—We should prefer to take the water from the heater to the feed pump, but in order to do this, it would be necessary to set the pump and heater so that water flows to the pump by gravity, as you could not lift the hot water by suction. It would "steam bind" the pump the minute any suction was put on the inlet chamber to the pump. Of course, if it is not possible to set the pump below the heater, then it will be necessary to pump through the heater, but we prefer the other arrangement.

209. Q.—How MANY gallons of water will a coil of 100 lineal feet of 1-in. copper or brass tubing heat per hour when surrounded by a temperature of 500 deg. F. the temperature of water entering the coil to be 50 deg. F. and discharged at 150 deg. F.? The tubing is to be of sufficient thickness to withstand a water pressure of 70 lb. per square inch.

Is there any rule or table for computing the number of gallons of water copper, brass or iron pipe will heat to a required temperature when the sizes and other temperatures are known?

A.—The method of finding the gallons per hour that can be heated under these conditions would be to multiply the area, in square feet, of the exterior of your heating coil by the average temperature difference between the water and the gases, and by a constant; divide this by 8.33, the pounds of water per gallon, and by the temperature rise of the water.

For 1-in. pipe, the length of 2.9 ft. will give a square foot of external surface, so that 100 ft. will give about 34 sq. ft. The average temperature of the water is 100 deg. F., and the difference between this and 500 will give 400 deg. temperature difference.

The constant for copper or brass pipe would be about 250, and the temperature rise of the water is 100 deg. Using these values, it would figure out to about 4000 gal. of water an hour which would be heated by 100 ft. of copper pipe.

The thickness of the pipe would make very little difference, as the chief resistance to the conduction of heat is at the surface of the pipe. For wrought iron or steel pipe, the constant will be practically 200.

210. Q.—A TANK with capacity of 1000 gal. of water is to be heated to 200 deg. F., using exhaust steam

from an 8 by 10-in. high-speed engine making 280 r.p.m. Boiler pressure is 80 lb., and the exhaust pipe 2½ in. About 500 gal. of water is used a day. What amount and size of pipe is needed in the heating coil, and where in the tank should the coil be placed? All water is taken from the bottom of the tank, which is 15 ft. from the engine and 6 ft. from the floor.

A.—The capacity of the tank would have comparatively little to do with the amount of coil surface required, as this would be governed by the amount of water used per hour.

You speak of using 500 gal. a day, and we assume that this is for 10 hr., so that you would require 50 gal. an hour, which at 8⅓ lb. to a gallon would be approximately 420 lb. of water to be heated per hour.

For heaters, a good rule is, that with wrought iron pipe, 210 B.t.u. will be transmitted per square foot of coil surface per hour for each degree difference between the temperature of the steam and the average temperature of the water. Assuming that your water comes in at 40 deg., and is heated to 200 deg., the average temperature is 120 deg., and the steam from the exhaust in your engines would be in the region of 200 deg., giving a temperature difference of 80 deg. Multiplying 210 by this gives 16,800 B.t.u., which would be transmitted per hour per square foot of heating surface.

The rise in temperature of the water is 160 deg., and you have to heat 420 lb. an hour. It requires 1 B.t.u. to heat a pound of water one degree, so that 420×160 will give the total heat required, or 67,200 B.t.u. Dividing this by the heat per square foot of heating surface gives 4 sq. ft. approximately.

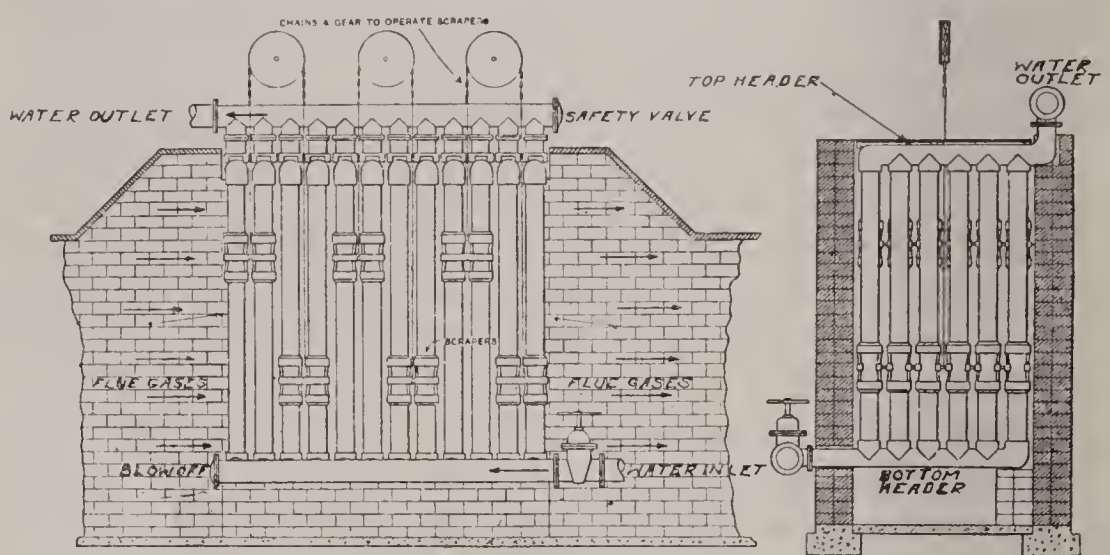
The pipe used would of necessity be 2½-in. pipe, as you do not wish to increase the back pressure on the engine unnecessarily. For this pipe it takes 1.3 linear feet to give a square foot of heating surface. Dividing 4 by this would call for approximately 3 ft. of pipe.

There will probably be times when you will want to heat at a rate much greater than 420 lb. an hour, as the demand is not likely to be uniform. We would, therefore, advise putting in 10 ft. of 2½-in. pipe, locating this 6 in. from the bottom of the tank, at about the center, which will tend to induce good circulation through the tank and keep the water throughout well heated. You realize, of course, that the hottest water

will be at the top, and for that reason you would probably get better results to take the water from the top of the tank, but we presume there are mechanical difficulties in the way.

211. Q.—How is an economizer constructed, and how connected to the boiler?

A.—The economizer is an arrangement of heating surface in the form of tubes, Fig. A211, placed between the boiler and the stack, for the purpose of raising the temperature of the boiler feed water. It consists usually of top and bottom headers, connected by tubes, the



A211. TYPICAL ARRANGEMENT FOR ECONOMIZER

water circulating in one or more passes, and ordinarily being sent to the economizer after having passed through the feed-water heater. The temperature is, therefore, raised to approximately that of the water in the boiler.

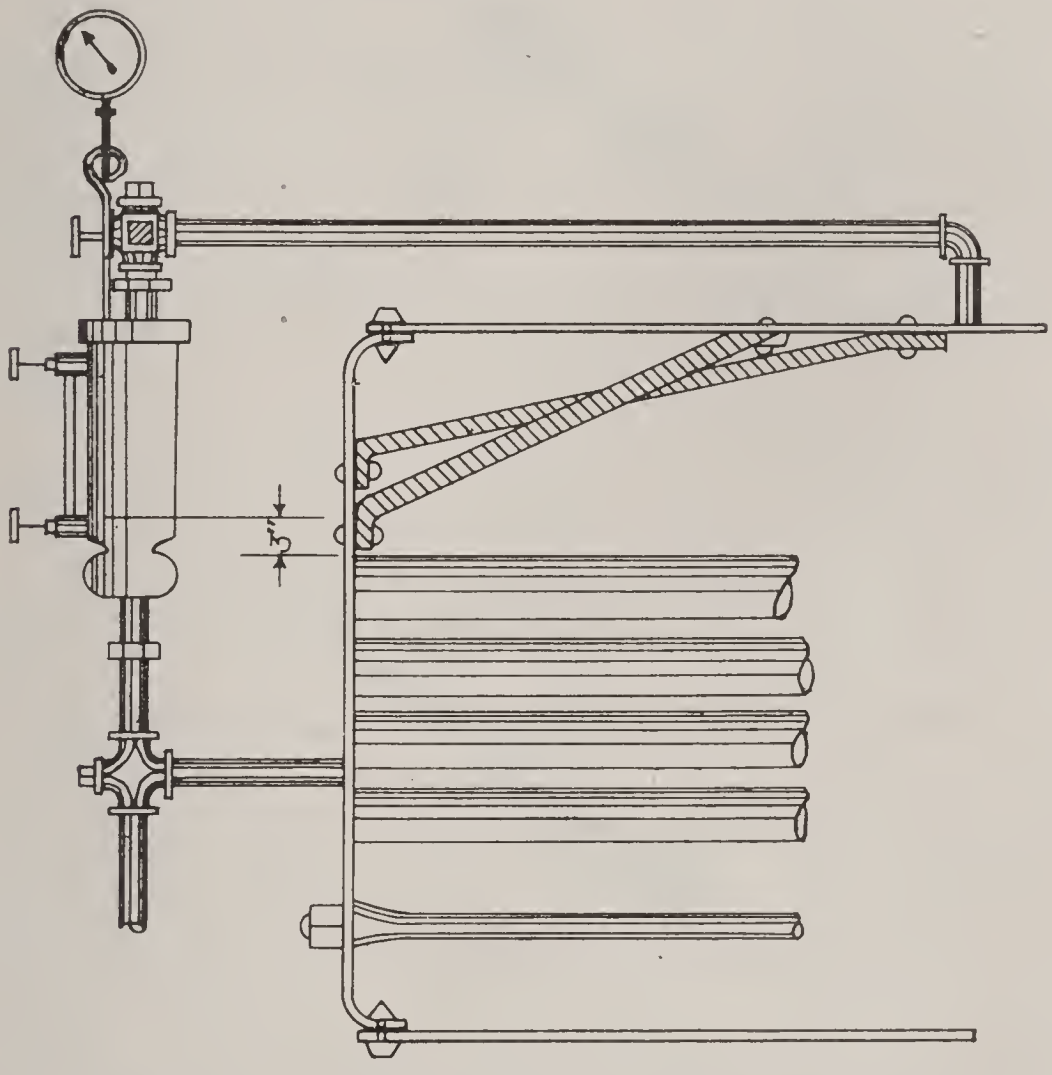
Discharge into the boiler is from the economizer through check and stop valves. An arrangement of scrapers which are continuously moved up and down on the tubes, these tubes being arranged vertical, is commonly provided to keep the tubes free from soot and ash, which otherwise accumulate rapidly because of the condensation of moisture on the outside of the tubes, due to difference in temperature between the tubes and the hot gases.

The economizer may be arranged in any position with reference to the boiler. Sometimes it is behind the boilers, sometimes on top, sometimes at one side. The uptake from the boilers to the stack has a bypass, so that

the gases may be sent around the economizer in case of repairs or renewals to the economizer being necessary.

212. Q.—What are the proper connections for water column and steam gage to the boiler?

A.—The proper connections for a water column should not be less than 1 in. in diameter for boilers under 54 in. in diameter, nor less than 1¼ in. diameter for boilers over 54 in. diam. The water column should be so placed that the bottom of the water glass is not



A212. CONNECTIONS OF WATER COLUMN

less than 3 in. above top of tubes. The steam connection should be piped directly to steam space of boiler, as in Fig. A212.

The water connection is piped directly to front head at a point considerably lower than the top row of tubes. No ell connections to be used and no shutoff cock used, except under extenuating circumstances when only a straight flow valve of the gate variety shall be used.

The steam gage should be connected to the steam space of a boiler by an independent line or may be connected to steam line connecting water column and boiler.

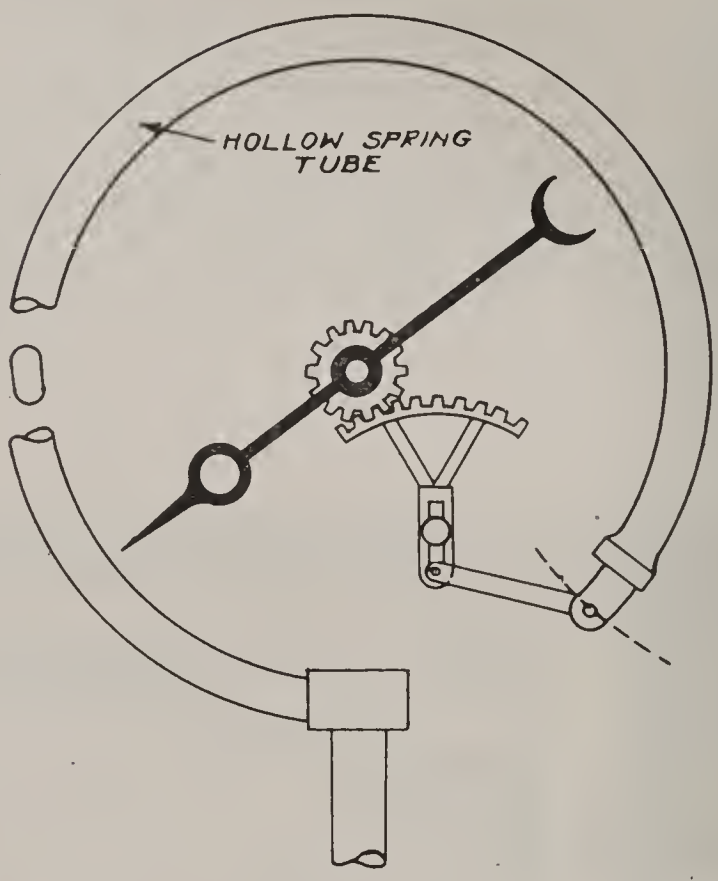


FIG 1

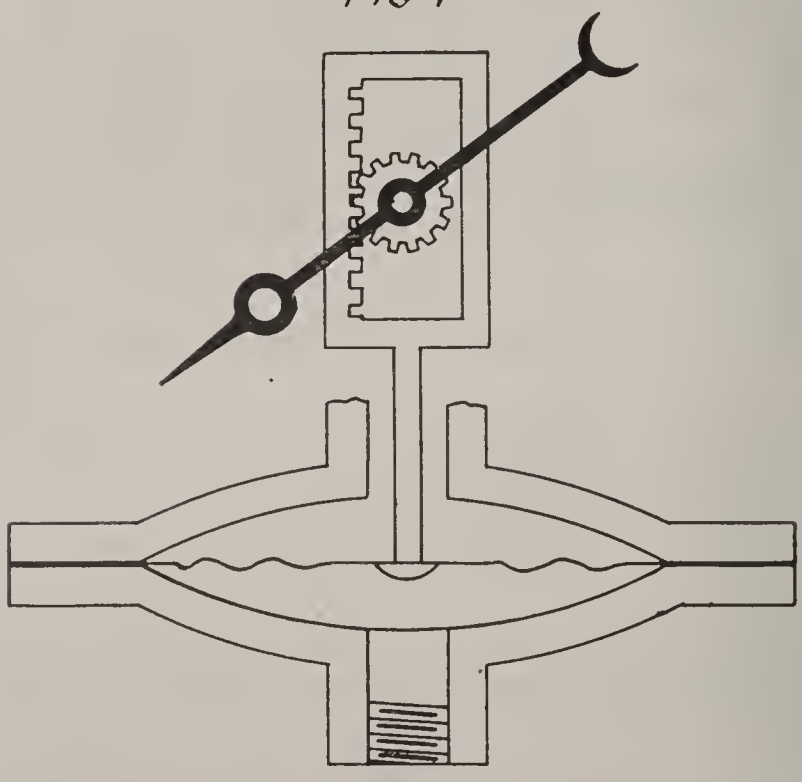


FIG. 2

A216-1. DIAGRAM OF STEAM GAGE

A216-2. CONSTRUCTION OF DIAPHRAGM GAGE

213. Q.—What kind of valves are allowed as shutoff between boiler and water column?

A.—The Ohio rules provide that shutoff valves between the boiler and water column must be straightway, outside screw and yoke locked open. No automatic shutoff is permitted in the gage glass fittings, unless gage cocks or other means of ascertaining the water level are provided independent of the water glass.

214. Q.—What precautions should be taken when putting in new gage glasses?

A.—When putting in a new gage glass, the precaution to be taken is to see that no unnecessary pressure is exerted by the stuffing box glands. It is also well to have the glass a trifle shorter than the distance between the bottoms of the stuffing boxes, so that as the glass expands due to the heat of the steam and water, it will not be placed under a strain between the bottoms of these boxes. The glass should not touch metal at any point.

215. Q.—What is the object of placing the gage cock outlets on an angle on the head of a steam boiler?

A.—Try cocks on a boiler are placed on an angle for two reasons—to direct the discharge away from the operator, and also from the cock immediately below.

216. Q.—Explain the working of pressure and vacuum gages.

A.—Ordinary steam gages consist, as shown in Fig. A216-1 of the so-called "Bourdon" tube, elliptical in cross section and circular in form. One end of this tube is fixed and provided with an opening for air or water; the other end is generally connected to a pivoted circular rack meshing with a small pinion mounted on the same shaft as is the pointer. Air or water is allowed to enter this tube, and when pressure is applied, due to its elliptical section, the tube will have a tendency to straighten out, the free end describing the path shown by the dotted line. This action, when communicated to the rack, causes the pinion to turn and likewise the pointer, the amount of turning being directly proportional to the pressure.

In gages of this type, it is essential that the steam never be allowed to come in direct contact with the tube,

for should this happen, the high temperature will damage the tube and render the gage useless.

Another kind of gage is one in which a metallic disc or diaphragm, instead of a bent tube, actuates the indicating device. It consists of a corrugated diaphragm, Fig. A216-2, clamped around its edge by the flanges of an encircling chamber. Pressure applied on the lower side of the diaphragm deflects it upward, the amount of this movement being proportional to the pressure. By means of a connecting strut, the movement of the diaphragm is communicated to a rack connected to a small pinion attached to the spindle of the pointer.

For vacuum measurements, Bourdon gages are very commonly used. The design of a pressure gage is altered only in the arrangement of the levers moving the needle, which for vacuum measurements turn the needle in the same direction as for pressure; but in this case, the tube is bent inward or toward the center of the gage instead of outward as for pressure measurements.

Another type, known as a compound gage, is used to indicate either pressure or vacuum on the same dial. The linkages are adjusted differently from those in the usual types, in that the position of the pointer for zero gage pressure would be at about the top of the dial.

217. Q.—What is the best way to test a steam gage?

A.—The best way to test a gage is on a gage tester, several types of which are on the market. If one of these is not available, put the gage on the same line with another gage known to be correct, vary the pressure in the line between the reading limits of the gage, and compare readings of the two gages.

Steam and Heating

218. Q.—What is a simple rule for estimating the flow of steam in a pipe?

A.—To estimate the flow of steam in pipes, the following rule has been found as accurate as any simple form:

$$W = C \times \sqrt{p \times D \div L}.$$

Where W = the weight of steam flowing in pounds per minute;

p = the drop in pressure through the pipe in pounds per square inch, due to friction;

D = the density of the steam, or weight of a cubic foot in pounds at the given pressure;
L = the length of pipe in feet, including the equivalent length added for valves, elbows, etc.;
C = the coefficient depending on the diameter of the pipe.

The constant C of this formula is for 1-in. pipe, 46; 1½-in., 159; 2-in., 320; 3-in., 977; 4-in., 2016; 6-in., 6210; 8-in., 13,050; 10-in., 23,605; 12-in., 38,074.

219. Q.—With two jets of steam 2 in. in diameter, 100 lb. initial pressure, one jet discharging into a vacuum, the other jet discharging against 75 lb. pressure, which jet will have the greater velocity? Why?

A.—The jet of steam discharging into the vacuum will have a somewhat higher velocity than that discharging against the pressure of 75 lb. This is due to the fact that the flow of steam of a greater pressure into an atmosphere of less pressure increases as the difference in pressure increases until the external pressure becomes only 58 per cent of the absolute initial pressure. The flow of steam is neither increased or diminished by the fall of the external pressure below 58 per cent or about 4/7 of the inside pressure, even to the extent of a perfect vacuum.

100 lb. gage is 115 lb. absolute; 75 lb. gage is 90 lb. absolute; which is 78.2 per cent of 115 lb., hence the velocity of this jet will be less than of that discharging into vacuum. Had this first jet discharged against a pressure less than 58 per cent of the initial pressure, there would be no difference in velocity between the two jets.

220. Q.—Does steam condense more quickly when in motion than when at rest?

A.—Steam will usually condense more rapidly when it is in motion in the pipe, because it shifts in its contact with the pipe, and a greater volume of steam comes in contact with the radiating surfaces. Condensation is due, of course, to the radiation of heat to the cooler air outside. When the steam is still, a comparatively small quantity comes in contact with the surface of the pipe. When the steam is moving, the quantity is increased, and the heat radiated increases.

221. Q.—To what temperature can superheated steam be raised?

A.—There is no limit except the mechanical of transferring heat to the steam.

222. Q.—Does superheating steam to a higher temperature increase its pressure?

A.—Where steam is superheated, its temperature alone, and not its pressure, is increased.

223. Q.—Can you use superheated steam in all types of engines? Why?

A.—A small amount of superheat can be used in all engines, but it is found in the Corliss and slide valve engines, that a large amount of superheat, say over 100 deg., is apt to make trouble from cutting of the valves, and also sometimes from poor lubrication in the cylinders. Wet steam carries lubricating oil readily. Superheated steam does not.

224. Q.—In a heating system supplied with live steam, to be changed to exhaust heating from an engine having no back pressure valve,—returns to discharge to an open heater, what changes besides separating the oil from the exhaust steam are necessary?

A.—In regard to the change in the heating system, it is not likely that there need be any changes, except, of course, that there will have to be some connection made from the live steam system to the exhaust heating system, to draw steam in case the exhaust from the engine is not sufficient. This connection must be fitted with a reducing valve, so that steam coming from the live system will never be at greater pressure than that which it is expected to carry on the exhaust of the engine. Aside from this, there is no change, except to put an oil separator into the exhaust line from the engine, so as to take out all possible oil and condensation before the exhaust steam goes to the heating system.

225. Q.—In vacuum steam heating, how is the vacuum established and maintained?

A.—Most of the vacuum systems in use employ some form of patented device for removing the air and for automatically maintaining a slight vacuum within the radiator. Any low pressure heating system may be transformed into a vacuum system, provided the joints

are sufficiently light, by substituting vacuum air valves in place of the usual automatic valves. These air valves are so designed as to allow the air to pass out of the radiators when the steam pressure within is slightly above that of the atmosphere. When the air is once expelled, the steam pressure may be allowed to drop through a considerable range, while the valve remains closed and does not allow the air to enter the radiator.

226. Q.—What size steam and return pipes should be used for heating coils 40 ft. long of 1-in. pipe?

A.—In regard to the size of pipe for connection to a steam coil, this depends on the surface in the coil. You speak of a coil of 1-in. pipe, 40 ft. long, but do not say how many pipes high. For a 1-in. pipe, it would take 2.9 linear feet to give a square foot of radiating surface, so that each 40-ft. length would give you 13.8 sq. ft. The size pipe required for radiators runs about as follows:

For	40	sq. ft.	radiation,	1	-in.	steam pipe,	¾	-in. return
For	75	sq. ft.	radiation,	1¼	-in.	steam pipe,	1	-in. return
For	126	sq. ft.	radiation,	1½	-in.	steam pipe,	1	-in. return
For	286	sq. ft.	radiation,	2	-in.	steam pipe,	1¼	-in. return
For	535	sq. ft.	radiation,	2½	-in.	steam pipe,	1½	-in. return
For	890	sq. ft.	radiation,	3	-in.	steam pipe,	1½	-in. return
For	1360	sq. ft.	radiation,	3½	-in.	steam pipe,	2	-in. return
For	2000	sq. ft.	radiation,	4	-in.	steam pipe,	2	-in. return
For	6000	sq. ft.	radiation,	6	-in.	steam pipe,	3	-in. return

227. Q.—How would you figure the required size of radiator for a given room?

A.—A simple and comparatively accurate rule to determine the required number of square feet of direct radiation is as follows: multiply the sum of glass surface and equivalent glass surface in square feet by the difference between the desired temperature of the room and that of the external air; divide this product by the difference between the temperature of the heating surface and the desired temperature of the room.

The equivalent glass surface referred to is obtained by dividing the exposed wall surface by 10.

When the building under consideration is of fair construction, add 25 per cent to the above, and if of poor construction, an additional 50 per cent of radiating surface is to be recommended.

228. Q.—How many pounds of steam at atmospheric pressure will be condensed in a cast-iron indirect

radiator to heat 25,000 cu. ft. of air from 15 deg. up to 120 deg. F.?

A.—We need, besides the data given, the specific heat of air at constant pressure, which is 0.2375, this being the number of B.t.u. required to raise the temperature of a pound of air one degree at constant pressure. We know, also, that a pound of air at 32 deg. F. and atmospheric pressure occupies 12.387 cu. ft. At 15 deg., the volume will be less in direct proportion to the degrees of absolute temperature. The absolute temperature of the Fahrenheit scale is equal to Fahrenheit temperature plus 461 deg., or 32 deg. F. is equal to 493 deg. absolute, and 15 deg. F. equals 476 deg. absolute. Multiplying the volume 12.387 at 32 deg., by 476, and dividing by 493, it gives us 11.96 cu. ft. as the volume of air at 15 deg. F. and atmospheric pressure. Dividing 25,000 cu. ft. by this gives 2090 as the number of pounds to be heated.

The temperature rise is from 15 deg. up to 120 deg., or 105 deg. Therefore, multiplying the weight of the air, 2090, by the temperature rise, 105, and by the specific heat, 0.2375, gives us 52,100 B.t.u. to be supplied by the steam. A pound of steam condensed at atmospheric pressure will give up 970.4 B.t.u., and dividing the heat to be supplied by this gives 53.55 lb. of steam to be condensed.

229. Q.—In changing the heating system of a four-room school building from hot air to steam—the dimensions of each room are approximately 28 by 32 by 10 ft. —what are the required number of square feet of radiation and size of boiler?

A.—In this school building, the amount of radiation needed in each room will be about 128 sq. ft. This may be supplied either in pipe coils or in cast-iron radiators. If the latter is used, it would require, say, four radiators of the two-column type, 26 in. high, in 14 sections, or about 35 in. long. It would be better to have more, rather than less, surface. The radiators should be set, so far as possible, underneath the windows.

For a plant of this size, the one pipe system will be satisfactory, and this would require a 2-in. supply pipe to each room, with a 1-in. connection to each radiator. It will probably arrange better to use two mains, each 3 in. in diameter, and each supplying two rooms.

These mains should be installed with a drop of 1 in. in 10 ft. towards the boiler, so that the condensed steam can drain back properly.

A cast-iron sectional boiler should be used, with capacity to supply 650 sq. ft. of radiating surface. This should have a safety plug in the section over the furnace, extending 1 in. inside the boiler. It should be supplied with steam gage, pop safety valve, water gage glass, and gage cocks. It should have returns from the radiators entering the boiler near the bottom, with check valves to prevent the water being forced back into the returns. There should be, also, a blowoff valve, and a water supply pipe. The water can be supplied from the city mains.

The mains in the basement should be covered with insulating material to prevent the waste of heat in the basement. For a plant of this size, this covers practically the requirements of the installation.

230. Q.—For a 35-room hospital building what kind of heating system would you recommend? Other buildings now in use are heated with exhaust steam from engines and pump in connection with a feed-water heater of the open type. The back pressure carried ranges from 1 to 5 lb. and the system is of the gravity return type. The heating returns from the present buildings are discharged into an open heater by gravity and pumped from there to the boilers.

The new building, which is about 300 ft. from the entrance of the boiler room, is so located as to be below both the feed-water heater and the return line.

Would you advise the installation of a vacuum system?

A.—The new building, being so low that the returns cannot be discharged into the heater by gravity, will need some arrangement to help the condensation up into the heater. This may be done as suggested in your letter, by installing a complete vacuum system in that building, or it may be arranged to keep the same system in that building as the others, by installing a lifting trap to take care of the returns. From the amount of information given, it would seem that this arrangement would be the most satisfactory, as the trap could be placed as low as necessary in the building, would be en-

tirely automatic in its action and would not require much attention.

A supply of medium pressure steam would be needed to operate the trap. A check valve on the vent pipe from the lifting trap would help some, effectually stopping any inrush of air which occurs after discharging by the condensation of steam left in the body of the trap. This would create a partial vacuum, which would naturally help to draw the condensation into the trap.

The vacuum system without any doubt would work well and all air valves would be eliminated, also smaller supply pipes could be used; but there would be pump troubles and annoyance from cylinder oil which would be needed to be used with the steam pump. If you have electricity which is on for 24 hr., then an electric, motor-driven vacuum pump may be used in place of the steam pump. The only drawback would be a higher first cost. With this system, radiator traps would be required for every radiator and coil, and an air separator would be needed for the discharge of the pump.

231. Q.—A ONE pipe steam heating system with cast iron boiler, and radiators is fitted with all necessary standard air valves. When raising steam the water level fluctuates badly, leaving the glass entirely at times. If any attempt is made to raise over 4 lb. of steam, the water appears to leave the boiler entirely. Water hammer is very severe in the system when first raising steam. Will placing a check valve in the return pipe at the boiler stop this, or will it be necessary to make the vertical drop of return pipes near the boiler greater and give more pitch to the return pipes?

A.—The only remedy for your condition is to change the piping so as to get a greater pitch to the horizontals, and as long a vertical drop as possible near the boiler. Putting in a check valve would have no value, as the water does not leave the boiler by forcing back up the returns. What does happen when you start to raise pressure is, that the water collected in the horizontals and pockets, and possibly in the bottoms of the radiators in your heating system, condenses the steam as rapidly as it is formed, in order to heat up this dormant water. Until the water is heated to the temperature of the steam, the new steam formed in the boiler will simply be carried on in the system and condensed, and little

back flow will take place from the system. The water does not leave your boiler entirely, but simply the level drops out of the gage glass.

That this action is what takes place is evidenced by the water hammer in your system, which means always the condensation of steam by contact with cold water, and a consequent surging and rushing of water in the pipes. You get this same noise when you discharge a jet of steam into a barrel of water, only that the noise is muffled, instead of being multiplied by the resonance of the piping system. Wherever you hear the water hammer loudest, will probably be the points where the water is settling, and these points should be given better drainage.

232. Q.—Live steam at 5 lb. pressure is carried in a dry house, the steam coming from a boiler run at 110 lb. through a reducing valve. Would there be any economy in fuel if the boiler pressure were reduced to 40 lb.? Would this make any difference in the dry house temperature?

A.—It certainly would make considerable difference in the consumption of fuel if you would maintain a pressure of but 40 lb. in your boiler instead of 110 lb., as you are now doing. So far as the difference in temperature of the dry house is concerned, there would be absolutely none, as the temperature of the steam corresponding to a pressure of 5 lb. would remain the same, under the condition, of course, that the pressure of 5 lb. would be maintained. Unless you are using the steam generated by the boiler for some other purpose than that of heating the dry kiln, it might be advisable to carry no more than 5 lb. on your boiler, in which case you could do away entirely with your reducing valve, and also effect a saving in fuel.

233. Q.—Please give an estimate of the amount of steam, and comments on the system, used in heating a brick factory building under the conditions stated below. Two fire-tube boilers having a normal rating of 55 hp. each and worked at a pressure of 100 lb. gage are used for the engine, heating and drying systems. The engine is of the non-condensing 4-valve type and consumes approximately 28 lb. steam per i.hp. hr. with a load varying from 90 to 115 hp. The heating system throughout the factory is in poor shape; returns are provided

for, but no vacuum pumping system is used, drips going to a well and pumped finally to the boilers.

From a rough survey approximately 2086 sq. ft. of 1-in. pipe is used for a factory having about 500,000 cu. ft. of surface. It has always been the custom whenever steam was needed for heating to use live steam at full boiler pressure, the system being fed from a 2-in. and 1-in. pipe. In the winter with the temperature anywhere below freezing, for the needs of the engine dry kiln and heating system about 8 tons of coal are used per day. The factory runs 10 hr. a day, dryer and heating system being on, to some extent, 24 hr.

What is the relative steam consumption of such a heating system using live steam and one as ordinarily used at say 15 to 20 lb.?

All things remaining the same, could one expect as much heat in the one instance as the other? When and why is use made of such a high pressure on a heating system? If a test showed that 40 to 50 per cent of the indicated engine load in the factory consisted of shafting and transmission loss, would not a saving be made by installing a low-pressure gas-fired boiler for the dry kiln and heating system, and so far as possible use individual motor drive?

A.—There is a certain loss of heat when heating with steam at high pressure due to the condensation being discharged at a high temperature from a trap or other device, and carrying away heat. At 100 lb. pressure, the temperature of the condensation is about 337 deg. F. and when in the hot well the temperature is only 212 deg. F. or less, dropping over 100 deg. F.; this heat evaporates some of the water in the discharge, which floats away as steam. The most economical pressure would be the lower or 20 lb., or even lower if possible.

Two thousand and eighty-six sq. ft. of direct radiation will use about 600 lb. of steam per hour, which is equal to 600 divided by 28, or 21.4 hp. at low pressure.

It is not stated whether exhaust steam is used in the dry kiln or not, but assuming it is not, it seems as if it would pay to make necessary changes in order to use it in both the dry kiln and heating system. It would be necessary to install a good oil separator in the exhaust pipe, and the hot well would need to be watched closely to make sure that oil did not get into the boiler.

On account of the steam being hotter, more heat will

be given off from the same amount of heating surface when using high pressure. The temperature of steam is dependent on the pressure, being 338 deg. F. at 100 lb. pressure and 260 deg. F. at 20 lb. If you should use a lower pressure it might be necessary to put in more piping for heating.

High pressure is used in heating systems more as a matter of convenience than anything else. If a system of piping is inadequate to heat the rooms where installed, the pressure is raised, thus making the pipes hotter and giving off more heat than before. Also, the condensation and air will be more easily taken care of, in fact, you can run your supply and return pipes almost anywhere and not have trouble as the pressure will force out the water and air.

The saving by buying electricity and using individual motors is not always as great as expected. If you can get rid of a great deal of shafting some saving will result, but even with the motors, there is some loss, as for instance, a small motor running light will have an efficiency of 50 to 80 per cent and then through the transformers and line there will be a 5 or 10 per cent loss, thus bringing the total loss about as much as with the present belt drive.

If you should put in a reducing valve for your heating line, you would make some saving. It is customary in many cases to carry 15 lb. back pressure on the engine and use the exhaust steam as far as it will go and then bleed into the system steam from the boiler through a reducing valve to keep up the supply, a high-back pressure being no loss if the steam is all used with the exception of loss of power in the engine which can be helped by raising the boiler pressure.

Boiler Efficiency

234. Q.—What are the chief causes of low boiler efficiency, and how may they be removed?

A.—The principal losses which tend to reduce the efficiency of a steam boiler are as follows: The first source of loss is caused by incomplete combustion; the second by inefficiency of the heating surface; the third source is due to radiation, while the fourth is latent heat of water formed by combustion.

Improvement in the efficiency of the boiler plant must be looked for from the following sources: By

improvement in the management of fires and the construction of the furnace the furnace temperature may be increased, the quantity of air required per pound of fuel may be reduced, and the loss due to incomplete combustion may be reduced. By careful arrangement in construction, and properly clothing the boiler with non-conducting material, you may reduce the radiation losses. Whenever the radiation loss is sufficiently reduced to warrant it, you can reduce the rate of driving by increasing the heating surface per boiler horsepower. The gas passages and heating surfaces may be so arranged that the gases are brought into thorough contact with the heating surfaces, thus increasing their conductivity. Heating surfaces should be kept clean both on fire and water sides to increase their efficiency. Water formed by combustion cannot be reduced to any extent, but preventing moisture in the coal as fired will help.

235. Q.—Analysis of coal used during a boiler test shows: Carbon, 80 per cent; hydrogen, 5 per cent; oxygen, 4 per cent; nitrogen, 2 per cent; ash, 9 per cent. Coal used, 4000 lb., and feed water 25,000 lb. per hour, temperature of feed 100 deg. F., boiler pressure 100 lb. absolute. What was the efficiency of the boiler?

A.—Dulong's formula for calorific value of 1 lb. of coal is:

$$\text{B.t.u.} = 14{,}600\ \text{C} + 62{,}000\ \left(\text{H} - \frac{0}{8}\right) + 4000\ \text{S}$$

B.t.u. = heating value in B.t.u. per pound of coal.

C, H, O, and S refer to the percentages of carbon, hydrogen, oxygen and sulphur respectively.

$$\text{Then, B.t.u.} = (14{,}600 \times .80) + 62{,}000\ \left(.05 - \frac{.04}{8}\right)$$

$= 14{,}470$ B.t.u. per lb.

The efficiency of boiler and grate equals, heat absorbed per pound of combustible divided by calorific value of 1 lb. combustible.

Heat in 1 lb. of steam at 100 lb. absolute pressure, from steam tables, equals 1186.3 B.t.u.

Feed water temperature being at 100 deg. F., the total heat supplied to 1 lb. of steam, under given condition is:

$$1186.3 - 100 + 32 = 1118.3\ \text{B.t.u.}$$

Since there was 9 per cent ash, the combustible per hour was, 4000 (1.00 — .09) = 3640 lb. combustible, in coal actually burned.

Water evaporated per pound combustible then equals

$$\frac{25{,}000}{3640} = 6.87 \text{ lb.}$$

The heat value of 1 lb. combustible equals

$$\frac{14{,}470}{.91} = 15{,}900 \text{ B.t.u.}$$

The efficiency of boiler and grate is

$$\frac{6.87 \times 1118.3 \times 100}{15{,}900} = 48.3 \text{ per cent.}$$

236. Q.—In a boiler test, 3000 lb. of water are evaporated at atmospheric pressure from feed water temperature of 160 deg. F. Coal used is 500 lb., which is 80 per cent combustible. What is the equivalent evaporation from and at 212 deg. per pound of combustible?

A.—From the total heat in 1 lb. of steam at boiler pressure, subtract the heat of the liquid at feed water temperature, and divide this by 970.4, which is the heat of evaporation at atmospheric pressure. The total heat at atmospheric pressure is 1150.4, and the heat of the liquid at 160 deg. F. is found by subtracting 32 from 160, which gives 128. This value subtracted from 1150.4 gives 1022.4, and dividing by 970.4 gives 1.053 as the factor of evaporation or pounds water evaporated from and at 212 deg. by the heat actually given to 1 lb. of water.

Five hundred pounds of coal, of which 80 per cent is combustible, would give $500 \times 0.8 = 400$ lb. of combustible. Dividing the 3000 lb. water by this gives 7.5 lb. of water actually evaporated per pound of combustible. Multiplying this by 1.053 gives 7.9 lb. equivalent evaporation per pound of combustible.

237. Q.—A boiler generates 100 lb. of steam per hour, having a quality of 98 per cent dry. If the gage pressure is 100 lb. and the temperature of the feed water is 100 deg. F., what is the equivalent evaporation from and at 212 deg.?

A.—It is first necessary to calculate the amount of heat absorbed by the water in the boiler.

One hundred pounds of water are raised from 100 deg. to 337.9 deg., which is boiling temperature at 100 lb. pressure. $(337.9 - 100) \times 100 = 23{,}790$ B.t.u. to raise the water to boiling temperature.

Latent heat of one pound of steam at 100 lb. pressure $= 880$ B.t.u. Since the steam contains 2 per cent of moisture, only 98 of tht 100 lb. delivered are steam, so $98 \times 880 = 86{,}240$ B.t.u. are used to convert the water into steam; the total then is, $23{,}790 + 86.240 =$ 110,030 B.t.u. absorbed by the boiler. The latent heat of steam at 212 deg. is 970.4. Dividing 110,030 by 970.4 gives 113.5 lb. nearly as the equivalent evaporation from and at 212 deg. F..

PART II

Engines and Turbines

Types of Engines

238. Q.—Name the advantages and disadvantages of a high speed engine.

A.—The advantages of a high speed engine are: It is well fitted for direct connection to an electric generator; compact; occupies small floor space; usually has a good system of lubrication; low first cost.

The disadvantages of this type of engine are: Comparatively high steam consumption and high rotative speed, resulting in greater wear and tear of the parts.

239. Q.—What advantages has a 4-valve engine over a single valve engine, and what are they? Is the Corliss engine the only 4-valve engine in use?

A.—The advantages of a 4-valve engine over a slide valve engine are as follows:

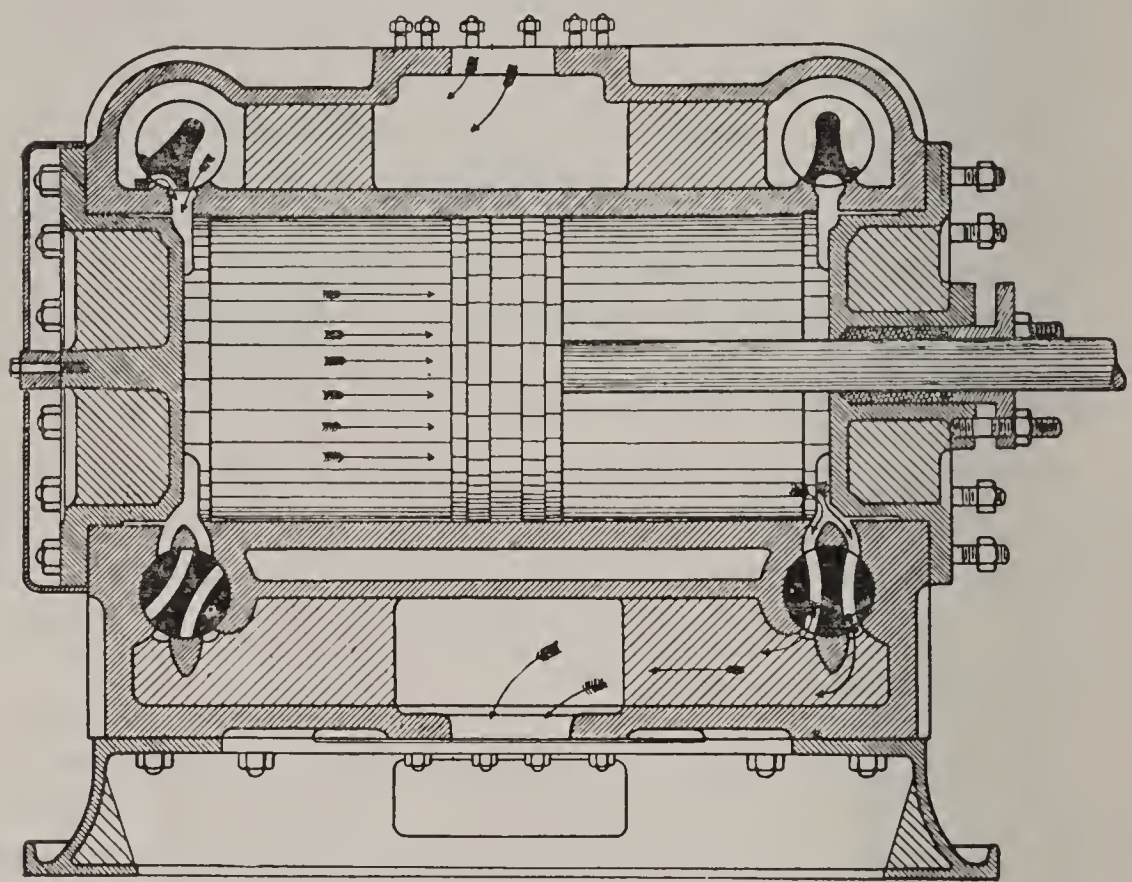
Good distribution of steam, resulting in better economy, due to the independent action of each valve relative to opening and closing. Cylinder drainage is almost perfect, owing to the position of the exhaust valves at the bottom of the cylinder, as in a Corliss type of engine. Full benefit of expansion of the steam, by admitting it at approximately boiler pressure and maintaining this pressure up to the point of cutoff, which is always automatic and proportioned to the load on the engine. This type possesses the advantage of being easily started or stopped and is easily manipulated.

The Corliss engine is not the only 4-valve engine in use. The Fitchburg engine is a 4-valve engine, and also the McIntosh & Seymour, which has flat grid-iron valves.

240. Q.—What five advantages has the Corliss steam engine over a common slide-valve engine?

A.—Five advantages of the Corliss engine over the slide valve engine are: (1) The valve ports may be made short and direct; (2) Clearance, volume and area are reduced to a minimum, resulting in a reduction of steam consumption; (3) Valves are made to operate promptly, without loss of power due to wire-drawing; (4) Closing of inlet valve at point of cutoff is effected in such a way

as to leave the points of admission, compression and release unchanged, which permits them to occur at the proper points of the stroke; (5) The ordinary slide valve has a heavy pressure bearing upon it throughout its travel, causing wear, not only of the valve, but also of the seat.



A240. SECTION OF CORLISS VALVES

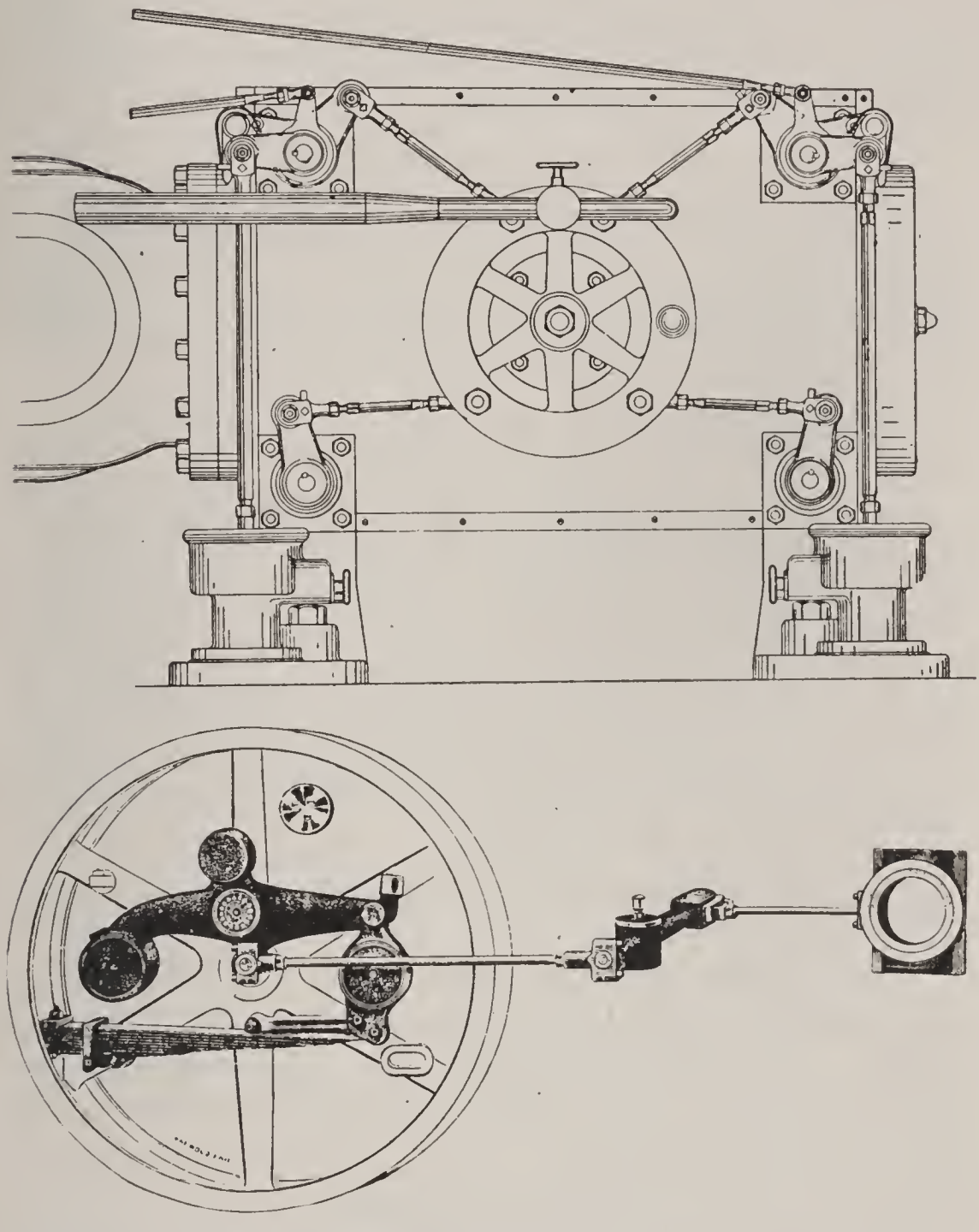
241. Q.—What is the most economical type of steam engine?

A.—The most economical type of steam engine depends entirely on the use for which it is to be installed. Generally a triple expansion engine, with high piston speed and automatic cutoff governor of some type will give the least steam consumption per horsepower-hour. If, however, exhaust steam is wanted for industrial use, it may be that a slide valve engine will give the most economical plant.

242. Q.—What is the difference between Corliss and automatic cutoff engines?

A.—The principal difference between Corliss and automatic cutoff engines is in their valve gearing, and the method of governing. In the Corliss engine, the steam valves and exhaust valves are operated by a fixed

eccentric on the shaft. These engines are governed by means of fly-ball governors, which, due to the variation in their position, cause the steam valves to close earlier or later, as the demand for power decreases or increases. Steam valves on a Corliss engine are usually closed by means of dash pots.



A242. DIAGRAMS OF CORLISS AND SHAFT GOVERNOR VALVE GEAR

In the automatic cutoff engine, the steam and exhaust valves are likewise driven by eccentrics mounted on the shaft. These eccentrics, however, are not rigidly connected to the shaft, as in the case of a Corliss engine, but are connected through springs to the flywheel gov-

ernor, so that as the speed increases or decreases, the eccentric will assume different positions, and thereby allow an earlier or later cutoff of the steam.

243. Q.—What is the difference between a single and double-acting engine?

A.—A single acting engine has but one impulse per revolution, and a double acting engine has two impulses, one on each stroke. In the single acting, pressure acts on only one end of the piston.

244. Q.—What is a square engine?

A.—A square engine is one in which the length of stroke and diameter of cylinder are the same; that is, a 10 by 10, or a 12 by 12. Usually the stroke is longer than the diameter, but for compound engines, frequently the low-pressure cylinder comes about square.

245. Q.—What is the difference between a condensing and a non-condensing engine?

A.—A non-condensing engine is one that exhausts directly to the atmosphere, or against a backpressure, as required by industrial conditions. A condensing engine exhausts into one of several types of mechanical devices called condensers, which not only condense all of the exhaust steam, but also create a partial vacuum which lessens the amount of steam required to produce a given amount of power, on account of the removal of back pressure on the piston.

246. Q.—What types of engines use governors that give variable lead?

A.—Engines having shaft governors which swing the eccentric around the shaft, give variable lead and cutoff. In some types, where the eccentric is swung across the shaft, the lead will be varied slightly, but not to any great extent.

Pistons and Rods

247. Q.—What is considered safe speed for an engine piston?

A.—The safe piston speed of engines varies for different types. For small stationary engines, the speed varies from 300 to 600 ft. per minute; for large stationary engines, it is about 750 ft. per minute; for marine engines, about 850 to 900 ft. per minute. The

best practical results are secured, ordinarily, with a speed of 600 ft. per minute.

248. Q.—Where in the stroke does the piston travel the fastest?

A.—The greatest velocity of a piston is between the head end dead center and the center of the piston travel.

249. Q.—Is there any difference between the angularity of the connecting rod of a horizontal engine and that of a vertical engine?

A.—There is no difference between the angularity of the connecting rod for horizontal and vertical engines, except that in vertical engines the ratio of the length of connecting rod to that of the crank is usually less, so that there is a greater angularity effect.

250. Q.—Why are some piston rings made eccentric? Should such rings be free to move in the slots, or be held in one position?

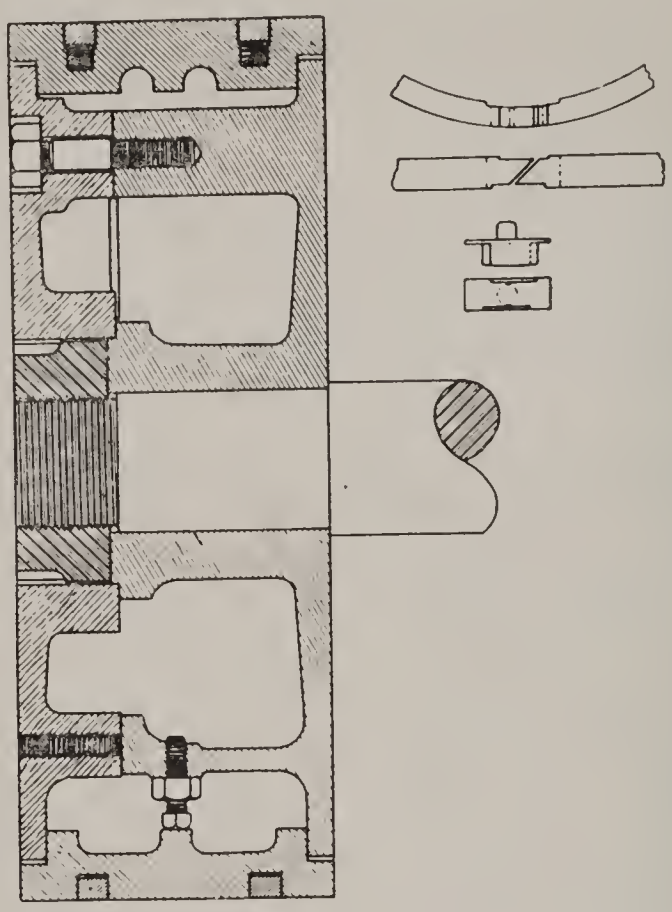
A.—Snap rings are made in two general types. In one type the ring is held in place by a dowel pin which prevents it from turning. In the other type, one side of the ring is made heavier, and the joints are located at different points in relation to the heavy side. This heavy side will remain on the lower side of the piston, thereby breaking joints in the rings. Preventing the ring from turning in the slot also has the advantage that the ring wears to the bore of the cylinder, which may not be exactly round, and will give less leakage than it would if allowed to turn. The ring should be a neat fit in the slot but not tight enough to bind.

251. Q.—Why are snap piston rings made thicker on one side? How are they measured?

A.—In case of a solid snap ring, for large engines the ring is made heavier at the side opposite the split, so that the ring when sprung into position will assume a shape more nearly round. If it were the same thickness all the way, there would be a tendency for it to bend more at the side opposite the split than around the rest of the circumference. A piston ring is dimensioned by the width and by the diameter of the cylinder.

252. Q.—How is a built up piston centered in the cylinder?

A.—In the built up piston, there is the solid part of the piston, then a bull ring, which is independent of the solid part, and can be centered by means of adjusting screws which bear on the outside of the piston body and on the inside of the bull ring. In this way, the piston may be centered in the cylinder by changing the location of the piston body with respect to the bull



A252. SECTION OF BUILT UP PISTON

ring. These adjusting screws are radial, and can be changed and fixed in position by jam nuts when the follower plate is removed. After the piston is centered, the piston rings are, of course, put in and the follower plate bolted on.

253. Q.—WHAT ADVANTAGE has a built up piston over a solid piston?

A.—The built up piston is usually lighter weight than the solid piston. It is easier to renew the rings and the center part of the piston can be adjusted to keep the piston rod always central with the gland.

254. Q.—HOW IS the size of piston rod figured for a 150-hp. engine?

A.—The size of the piston rod is determined, not by the horsepower of the engine, but by the maximum un-

balanced pressure to which the piston is subjected. One formula for figuring the size of steel rods is:

$$d = 0.0144D \sqrt{p}$$

where d is the diameter of the piston rod in inches; D, the diameter of the piston in inches, and p, the maximum unbalanced pressure on the piston in pounds per square inch.

Assuming a piston 14 in. in diameter, under 150 lb. maximum pressure, and substituting, we have:

$$\begin{aligned} d &= 0.0144 \times 14 \sqrt{150} \\ &= 0.0144 \times 14 \times 12.25 \\ &= 2.47 \text{ in. diameter of piston rod.} \end{aligned}$$

255. Q.—What is the best method of securing a piston rod to a cross head? And also to a piston?

A.—The best method of securing the piston rod to the crosshead is by screwing the rod into the crosshead and holding it in place with a jam nut. To secure the piston to the rod, the best means is a taper fit of the rod into the piston, and a nut on the end of the rod, this nut being held either by a locknut or by a setscrew half in the nut and half in the rod.

256. Q.—At what part of the stroke is the pressure greatest on the crosshead guides?

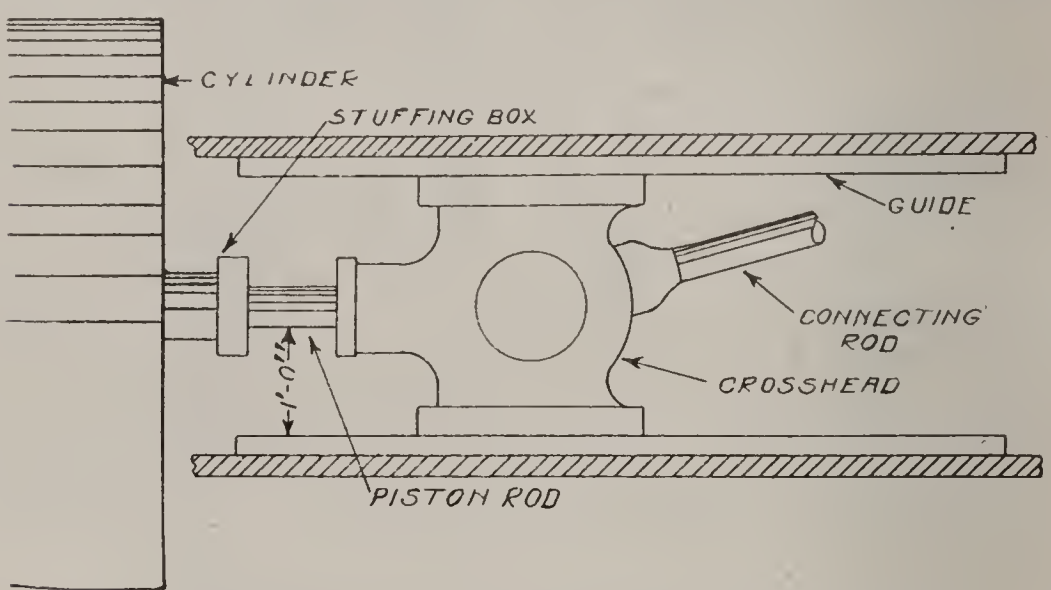
A.—The pressure on the guides in an engine is usually the greatest just before cutoff. The pressure is produced by the side thrust of the connecting rod, and will be greatest when the angle of the rod with the center line is greatest, and when the difference between the pressures on the two sides of the piston is greatest.

Just before cutoff, there is nearly boiler pressure on one side, and exhaust pressure on the other side of the piston, so that there is the maximum of pressure difference; and with cutoff at the usual point—about ¼ stroke—the angle of the connecting rod is nearly at its maximum. If cutoff occurs very early, say at 1/10 stroke, the increase of the pressure on the guides, from the increase of the connecting rod angle, would be greater for a time after cutoff than would the fall in pressure on the piston due to expansion. If, on the other hand, cutoff was at ½ stroke, the angle of the connecting rod with the center line would be nearly maximum before cutoff was reached. The angle which the connecting rod will make with the center line depends on the ratio of

the length of the crank and the length of the connecting rod, and all these things must be taken into consideration in determining the point at which the guide pressure is greatest.

257. Q.—How would you center the piston in a small engine?

A.—Turn the flywheel until the piston head is at the head end of the cylinder. Then measure the height, Fig. A257, of the piston rod, at a point just outside



A257. MEASUREMENT FOR CENTERING PISTON

the stuffing box, with reference to some fixed point below it. Now turn the flywheel till the piston head is at the other end of the cylinder. From the same fixed point as before, measure the height of the part of the piston rod that is now just outside the stuffing box. Raise or lower the crosshead until the two heights are the same.

258. Q.—In centering a piston, how would you proceed if there was no centering mark on the end of the rod?

A.—If there were no center mark on the end of the piston rod, the simplest thing would be to take a centering square and find the center. If this cannot be done, you can caliper from the outside of the rod to the bore of the piston, or you can use a thin wedge between the outside of the piston and the bore of the cylinder, centering the piston so that this wedge goes in an equal distance between the piston and cylinder at all points on the circumference.

259. Q.—How would you proceed to put in a new set of follower bolts in an old piston?

A.—Followers are attached to the piston by one of three methods: Tap bolts, ordinary studs, and safety studs. Each one of these methods differs somewhat from the others and not knowing which one is referred to in the inquiry, to be on the safe side, the process is given for all of them.

We will presume that the follower is held in place by tap bolts and that these have become worn so that it is necessary to renew them. With a new tap (if possible, or at least nearly new) carefully tap out the holes in the piston so as to insure uniformity. If one or two are enlarged considerably, use a tap of, say, a sixteenth inch larger in size for these. If so desired, to secure uniformity of size, all the holes can be re-tapped.

Select a good tough grade of machine steel, and of the proper size "hex." for the new tap bolts, and with a screw machine or screw cutting lathe, make the new screws just tight enough so that they can be screwed into place in the piston with a little strain on the wrench, but not so loose that they can be screwed in by hand.

Cut the head at least the same thickness as the depth of the counterbore, or a small amount less, and when screwing the tap bolts in, use cylinder oil and a little graphite flour as a lubricant.

Follow the same method of preparation for studs, either common or "safety." But in any case, should it be necessary to drill out a broken stud or tap bolt, use great care to make a good job of it.

Chip off uneven edges of the broken stud so as to present a fairly even surface to the drill. With dividers, locate the exact center of the broken stud and mark plainly with a center punch. Then drill out the stud, using the same size drill as would be used to drill the original hole. If the stud be large, say, inch or over, it would be best to drill the center of the stud with a small drill of a quarter inch or so.

In making the studs, use good tough machine steel, or cold-rolled if handy and preferred—only see that the fit of the tight end is such that it will require some strain to screw it into place. For a stud 1 in. in diameter, the fit ought not to be so close but that the

stud can be screwed into place with a 24-in. wrench. Other sizes in proportion.

The loose end of the stud ought to be the same fit as has been described for tap bolts, so that the nuts will

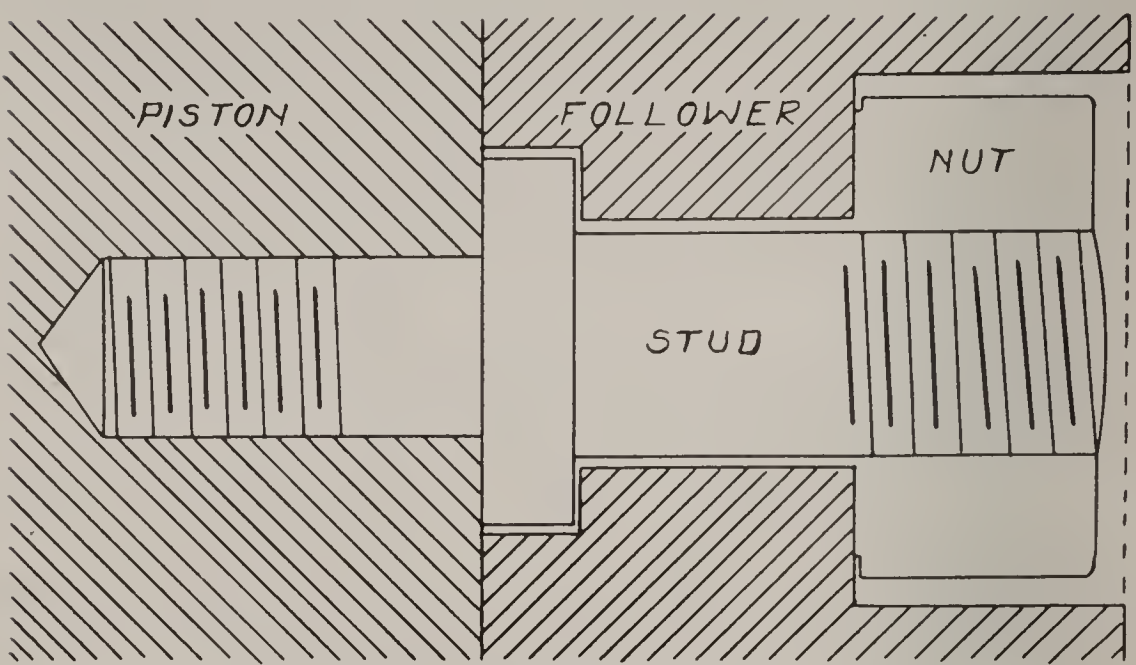


FIG. 1

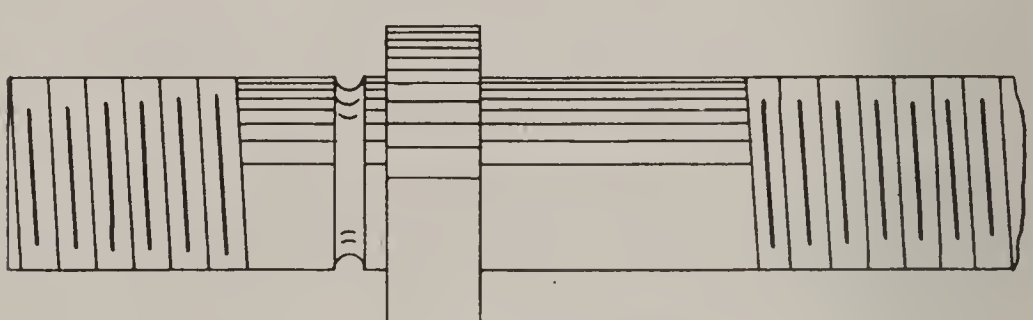


FIG. 2

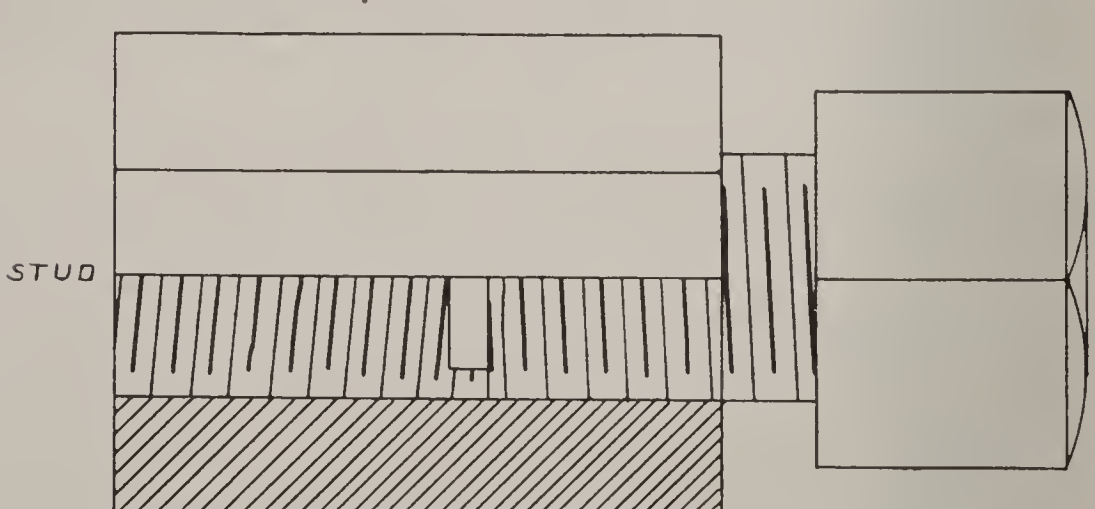


FIG 3

A259-1. SAFETY STUD FOR FOLLOWER
A259-2. GROOVED NECK SAFETY STUD
A259-3. DEVICE FOR SCREWING IN STUDS

require a common wrench to draw them up the entire distance.

The counterbore in the follower to receive the head of the tap bolt or the nut of a stud, should be at least the same depth as the diameter of the bolt or stud, as, for instance, for a 1-in. bolt or stud, the counterbore should be not less than 1 in. in depth.

The safety stud referred to has some features which recommend it for almost every condition, especially where the service is severe in large units, such as railway and rolling mill service. The stud is usually made with the two ends of unequal size and with a small collar in the middle, as shown in Fig. A259-1.

The idea is that if the stud should break, which is liable to happen when not expected, the fracture will occur at the weakest place, which is at the neck of the tight end which is screwed into the piston. The small collar then prevents the broken part from dropping out into the cylinder where it would cause considerable damage. Where the safety stud is used, the necessary strength is calculated at the smaller end.

A type of safety stud that has been used to a limited extent is the same size at both ends, Fig. A259-2, but having the collar the same as the one described above. The safety device consists of cutting a groove at the base of the collar on the piston side where fracture will occur first. The smallest diameter is used as the basis of calculation.

In screwing studs into place, do not by any means use the time honored pipe wrench. This mars the stud and in many cases the thread as well, and besides is hard on the pipe jaws. Use a stud nut. If none are at hand, and the thread on the stud will permit, use two common nuts jammed together tight enough so that the strain will exceed the friction of the entering thread. Use a good wrench on the outside nut and to release, transfer the wrench to the inside nut.

A good stud nut, Fig. A259-3, is easily made by taking a piece of round or hexagon steel the size of the nut for the stud in question, and about 2½ times as long as the diameter of the stud. Drill it through endwise and tap it. Then in one end screw a short tap bolt or a dog point setscrew (or even a common bolt if nothing else can be made or found quickly) so that the end will pass about half way through.

Start this on the stud and screw the stud in by means of a wrench on the setscrew and when on far enough, reverse the strain on the setscrew and the stud nut when it will come off easily.

260. Q.—If you were ordering a new set of piston rings, how would you obtain the proper size?

A.—Measure the diameter of the cylinder and have the rings turned, before being cut, about ¼ in. diameter larger than the cylinder, for cylinders up to 20 in.; then have enough cut out of the rings to spring them to the diameter of the cylinder. For larger cylinders, the rings will be cut proportionally larger.

The thickness is commonly made 1/30 of the diameter of the cylinder, plus ⅛ in.; the width may equal the thickness plus ⅛ in. Mean thickness of an eccentric ring may be the same as for a uniform ring, with minimum thickness ⅔ the maximum.

Authorities differ to some extent as to the proper dimensions of piston rings; but the above, given by Kent, may be accepted for all practical purposes. Where you are replacing old rings, we would, of course, advise following the dimensions of these as closely as possible. In such case, the required diameter and thickness will be determined as above, but the width should be the same as that of the old rings. It may also be necessary to take into consideration the width and depth of space provided in the body of the piston to receive the rings.

261. Q.—Is it good engineering practice to round off the sharp corners, or edges, of piston rings, either on new rings or those which have been in use?

A.—In regard to rounding off sharp edges of piston rings, we do not know of any engine manufacturer who does this, and would not feel that it would be wise to round them off to any great extent; but it is considered good practice to break the sharp corner sufficiently to get rid of the knife-like edge which tends to scrape the oil off the cylinder walls. In cases where there is no particular difficulty in getting the cylinder to take lubrication properly, rounding off the corners is probably not worth while; but if there is any difficulty in getting the oil to stick to the cylinder walls, it is worth doing, as in a number of instances of which we have known, this has helped considerably in getting satisfactory lubrication.

262. Q.—What check should be made on new piston rings before putting them into the piston?

A.—Before putting the rings in, they should be tested in the cylinder to make sure that they are the proper diameter, and also should be tested in the grooves to make sure that they are a free fit sidewise, but not too loose. This can be done by putting the ring into the groove and rolling it around to be sure that there is no binding at any point of the ring.

263. Q.—How is a new set of snap rings put into the piston?

A.—To put a new set of snap rings on a solid piston, it is necessary to take the piston out of the cylinder, of course, then the rings are opened up sufficiently to let them pass onto the piston, and are pushed along until they slip into the grooves. If it is necessary to pass a ring by a groove, this may be bridged at a number of points with strips of thin sheet iron, so that the ring will slide over to the groove beyond without dropping in. The same method may be employed to remove the old rings; that is, spread them out and slip thin strips of thin metal underneath, so that the rings will be held out of the grooves and may be slid off the piston.

264. Q.—How would you put the rings in a built-up piston without taking the piston out?

A.—In a built up piston, there is the solid part of the piston, then piston rings, usually of the sectional type set out by a coil or some kind of an elliptical spring, then a bull ring, or junk ring, which is adjusted with respect to the body of the piston by means of set screws, another set of piston rings, and finally a follower plate, which is bolted to the head end of the piston.

Of course, the follower plate can be taken off, also the piston rings taken out, the set screws loosened and the junk ring removed, and the second set of piston rings taken out, without removing the piston from the cylinder, simply by taking off the head end of the cylinder and working through the clearance space. The first piston ring is then put in place, the junk ring slipped in and adjusted by the set screws until the piston rod center calipers to be in the center of the cylinder, using a pair of hermaphrodite calipers for this

work. Next put in the other piston ring, and put in the follower plate.

265. Q.—How should one go about setting up on worn crosshead shoes of a horizontal Corliss engine to keep the crosshead centered between guides?

A.—If your engine is level and in alinement, the piston rod will be level and the same distance from the crosshead guides at all points. If this is the case, and you wish to take up on the crosshead shoes, put the engine on crank end center and a mechanic's level on the rod, and take up evenly on the shoes. Any variation of the level will indicate that one shoe is being taken up more than the other. It is likely that the lower crosshead shoe and guide will be worn more than the upper, so that the crosshead end of the piston rod will be lower than the piston end. In that case, take up on the lower shoe until the rod is level, then check up the alinement of the piston rod with the crosshead guides, and if correct, take up on upper shoes, maintaining the piston rod level as indicated by the mechanic's level.

266. Q.—With crosshead pin worn oval, where should the crosshead be placed when keying up?

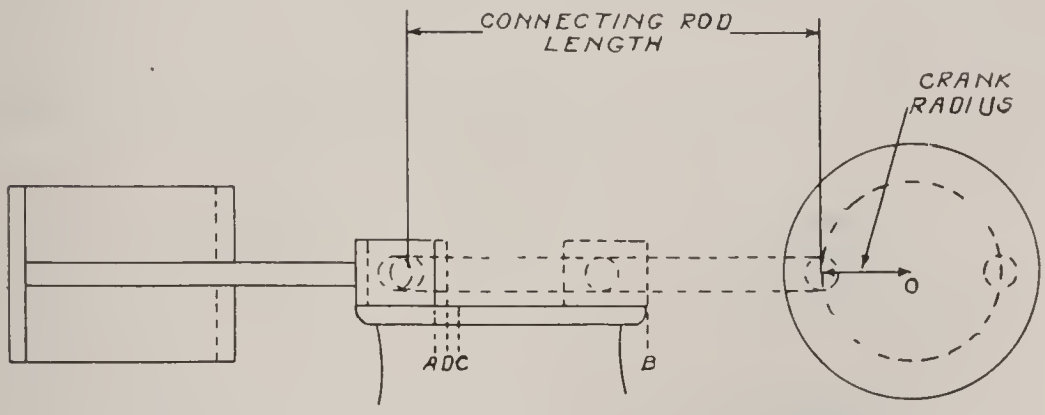
A.—If the crosshead pin had worn oval, the high parts would be on the top and bottom sides (in a horizontal engine) and, the boxes could not be keyed up very tight, as they would heat when coming up onto this shoulder. In order to key up without danger of heating, the boxes should be on the largest diameter of the pin. This would be at about one-half stroke. In a great many engines, however, the only point at which the wedge bolts can be turned is near the end of the back stroke. This would bring the boxes on the smallest diameter of the pin. In this case either file the shoulders off or turn the pin one-quarter around so that the wear will come on the largest diameter of the pin.

Connecting Rod

267. Q.—How would you measure the length of connecting rod required, neglecting the old one?

A.—To find the length of a connecting rod, place the piston at the extreme head end and at one end of the crosshead make a mark on the side of the guide, as A, Fig. A267, then move the piston to extreme crank

end and again mark the position of the same end of the crosshead as at B. Now lay off on the distance between these two marks a distance equal to twice the crank radius as BC. This will leave a small distance as AC which is the total clearance. Now bisect this distance, which gives the point D. Next place the end of the crosshead over the point D and place the crank at its



A267. FINDING LENGTH OF CONNECTING ROD

extreme head end position and the distance measured from the pin of the crosshead to the crankpin will give the connecting rod length.

268. Q.—In what position should the crank-pin be when setting up the shoes of the crosshead?

A.—The least wear on the crosshead guides is at the end of the stroke, so when setting up on the shoes, the engine should be on dead center, preferably on the crank end, as it is usually easier to get at it in this position.

269. Q.—In keying up brasses on connecting rod of the Fitchburg engine, what effect does it have on the clearance?

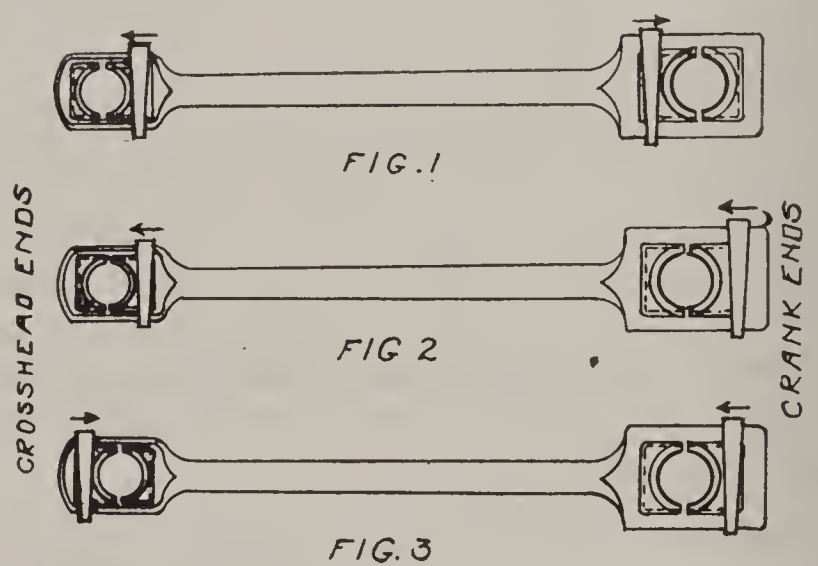
A.—The connecting rod of the Fitchburg engine has solid ends, and the keys are so placed that they draw the inner boxes toward the ends of the rod. Keying up will, therefore, increase the distance between cnters of crankpin and wristpin, lengthening the connecting rod, thus increasing the clearance at the crank end and decreasing clearance at the head end.

270. Q.—What effect does keying up strap-end connecting rods have on the length of the rod and on the cylinder clearance?

A.—Figure A270 shows three forms of what is generally known as the "plain key strap-end rod," in which the strap is slipped over the end of the rod and fastened by one or two bolts.

In A270-1 driving the keys will cause the boxes indicated by the arrows to move in the direction in which the arrows point and if either or both are driven, the effect will be to lengthen the rod and decrease the clearance at the head end of the cylinder.

In A270-2 driving the keys will cause the boxes indicated by the arrows to move in the direction in which



A270. FORMS OF STRAP-END CONNECTING RODS

the arrows point. Driving the key at the crank end will increase the clearance at the head end and driving the crosshead key will decrease that clearance. If the wear is equal on both boxes, keying both of them will not affect the clearance.

Driving the keys of the rod in A270-3 will cause the boxes indicated by the arrows to move in the direction in which the arrows point. Driving the key at the crank end will increase the clearance at the head end and driving the crosshead key will have the same effect, although the arrangement is not practical except for an engine with bar guides and an open crosshead.

271. Q.—How can the effect on clearance of keying up connecting rod ends be eliminated?

A.—To even up clearance, when the brasses become worn, place shims behind the brasses so as to keep the length between centers of crosshead pin and crank pin always the same.

Cylinders

272. Q.—How is the required thickness of steam engine cylinder walls determined?

A.—The thickness of the cylinder walls of an engine may be found from either of the following formulas:

A. $t = 0.0002\ p\ D + 0.4$ in.;
B. $t = 0.0003\ p\ D + 0.375$ in.

Where t = the thickness of cylinder walls in inches;
p = the maximum steam pressure in pounds per square inch, gage;
D = the diameter of cylinder in inches; if p is less than 100 in the actual case, use 100 in the design.

Formula A applies only to engines in which lightness is the prime consideration, and in which first-class material and workmanship are insured by careful inspection, and tests throughout the whole manufacturing process.

Formula B applies to cylinders above 10 in. in diameter, and is generally used for ordinary cylinders of slide valve engines.

273. Q.—What is the object of the counterbore in the cylinder?

A.—Steam engine cylinders are counterbored for the purpose of providing an edge over which the piston ring may pass, and thus prevent the wear from forming a shoulder in the bore of the cylinder.

274. Q.—How is the required cross-section area of steam ports figured?

A.—The area of steam ports of an engine should be such that the velocity of steam in passing through is about 6000 ft. per minute. The ratio of port area to cylinder area is in proportion to the piston speed of the engine. For a steam velocity of 6000 ft. per minute, the port area equals the square of the cylinder diameter times the piston speed in feet per minute, divided by 7639.

275. Q.—What may be the cause of a cylinder wearing on the bottom near one end?

A.—If the guides are not straight, or allow a considerable play of the crosshead at each end, excessive wear will result in the cylinder at the ends. The guides

should be straightened or made to fit closer to the crosshead. This will eliminate the greatest cause for excessive wear at the ends of the cylinder.

276. Q.—If the cylinder of a Corliss engine shows uneven wear, what is the cause, and what is the remedy?

A.—There are several reasons for uneven wear in engine cylinders and some others that are theories, only, but which seem to have substantial grounds for consideration. One reason may be a difference in the hardness of the metal in different parts of a cylinder, which naturally has its inception in the foundry.

If a casting be made from two different ladles of iron, or steel mixture, there is great liability of unevenness in texture. This can readily be seen when the cylinder is being bored at the shop, by the nature of the chips. Some cylinders are cast on the side and some on the end, so it can readily be seen what a difference in metal will do in creating a different texture of the iron in several places in the cylinder. If a cylinder has a hard spot, it will naturally wear less there than at other places.

Pistons having steam set packing rings will cause uneven wear as the pressure of the ring against the cylinder walls will be greater during the time of high pressure beginning with admission and continuing to the point of cutoff. From this point the wear is less till the mid stroke is reached, when the same peculiarity is noted on the other end. The shape of the cylinder is somewhat like a huge hour glass—small in the center and larger at the two ends.

Even pistons with snap rings, or with spring set rings have a tendency to wear the cylinder large at the ends as there is more or less pressure under the ring, especially after the engine has been in operation several years and the rings have worn enough to admit steam alongside of the ring into the recess under the ring.

If the lubricator be connected so that the oil will run down the inside of the pipe instead of being led into the center of the pipe by means of a "spoon" pipe where it will spray evenly, one side, or one end of the cylinder, as the case may be, will receive more than its share, and the opposite, proportionately less.

A case came under the writer's observation where the lubricator admitted the oil so that it would run

down the inside of the pipe and pass to the head end in preference to the crank end.

This was noticeable in the amount of oil on the surface of the cylinder walls at the two ends. The micrometer showed that the well oiled end was an average of 0.01 in. smaller than the other end, owing to better lubrication.

The inrush of steam into the cylinder will sometimes throw the piston to the opposite side, thus affecting the wear more or less.

These are the most important causes of uneven cylinder wear and the proper remedy is to remove the cause, if the wear is not too bad. But if it is, reboring is the only recourse, when naturally one would reassemble the cylinder so that the trouble would not occur again.

277. Q.—What is a simple way of finding cylinder clearance?

A.—The mechanical or piston clearance may be found by disconnecting the rod from either the crank or crosshead, pushing the piston against the head, marking the guide at the end of the crosshead, then pushing it against the other head and again marking the guide. These marks give the striking points. If the rod is now connected and the crank placed on the center, the distance from the end of the crosshead to the nearest mark will give the piston clearance for that end of the cylinder. With the crank on the other center the clearance for the other end may be found in the same way.

If the volumetric clearance is desired, place the crank on the center and fill the clearance space with water or cylinder oil to the face of the inlet valve. The amount required will give the volumetric clearance provided that neither the exhaust valve nor the piston leak.

The number of gallons required to fill the clearance, multiplied by 231, will give the cubic inches in the clearance.

When filling the clearance, the exhaust valve must, of course, be closed, and it is usual to fill the clearance through the port, if possible. In case the ports are on the side, the valves must be set so as to close both steam and exhaust ports and the filling is done through the indicator plug, if that is located on top of the cylinder. If the construction of the engine is such that this method

cannot be used, the only way is to measure up the clearance distance and the ports as well as possible and figure from this the cubic contents of the clearance.

Another method which may sometimes be employed is to insert a piece of soft metal, such as a piece of heavy fuse wire, or small stick of solder, in the indicator piping hole at either end and pinch the engine over the center. This will squeeze the soft metal to the exact amount of the clearance.

Engine Valves

278. Q.—How many working edges has a common slide valve?

A.—Four. Two admission and two exhaust.

279. Q.—How are different types of piston valves kept steam tight?

A.—Piston valves of the plug type, that is, with a solid piston, are kept steam tight by reboring the valve seats and putting in slightly larger valves, as wear occurs. In some types there are bushings which may be renewed as they wear, and in other types the piston valves have rings, the same as pistons, and can be kept tight by putting in new rings as the old ones wear.

280. Q.—Would a piston valve taking steam at the center be termed an "indirect" valve?

A.—A piston valve taking steam at the center would not be an indirect valve, but it would be a valve with inside admission. So long as the valve moves at all times in the same direction as the eccentric rod, it is a direct valve.

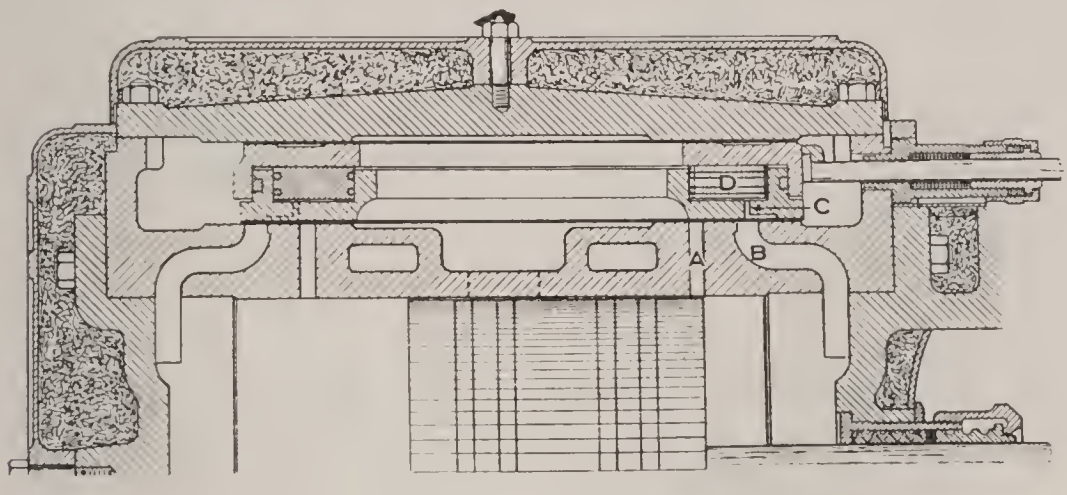
281. Q.—May an ordinary slide valve be termed a balanced valve? What is the difference between balanced and unbalanced valves?

A.—A common slide valve is not a balanced valve. Some slide valves are provided with plates whereby the pressure is equalized on both sides of the valve. This is frequently done by using a flat valve and a cover plate resting on side strips, the valve running steam tight between the seat and this cover.

282. Q.—How would you balance a slide valve?

A.—This cannot be done without change in the construction of the valve. The most common method is by the use of a ring on the back of the valve which bears

against a pressure plate. This ring is set into a groove in the back of the valve, and is forced out against the pressure plate by springs. Sufficient opening is made between the under side of the valve and the chamber within the ring to equalize the pressures, so that instead of full pressure forcing the valve against the seat, there is only pressure on that part of the top of the valve not covered by the ring. This is not a complete balance, but generally serves the purpose. A piston valve is completely balanced.



A282. BALANCING A SLIDE VALVE

283. Q.—Which is in the lead, the eccentric or the crank?

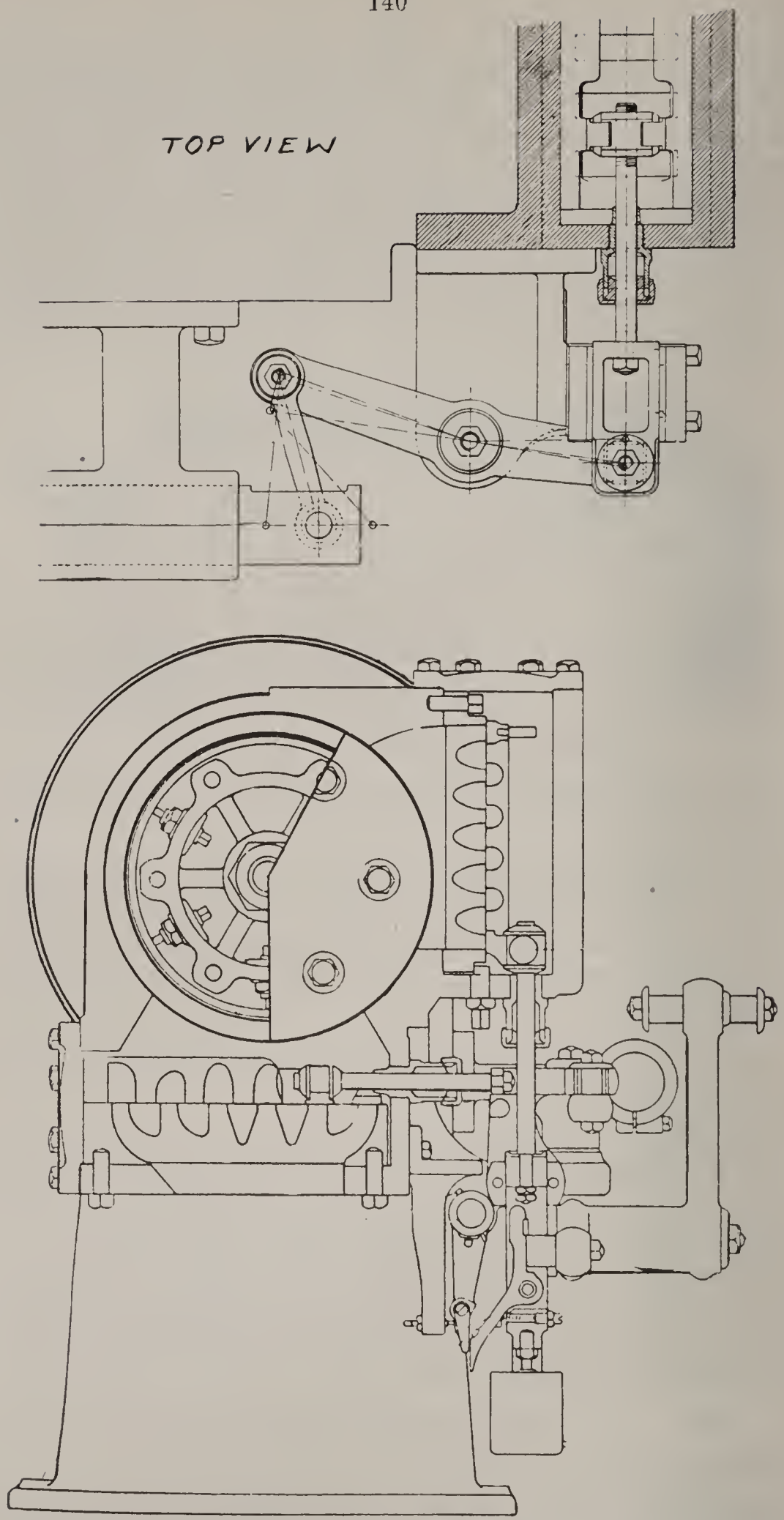
A.—In a direct-motion engine, the eccentric is ahead of the crankpin. With an indirect motion, the eccentric follows the crankpin. Both statements apply to a valve which takes steam at the ends of the valve and exhausts on the inside.

284. Q.—How high a lift has the poppet valve on the Putnam engine?

A.—In a Putnam engine the steam valve at the head end is given a lead rise of 1/16 in., while the steam valve at the crank end is given but 1/32 in., for the reason that the piston travels much more slowly in the crank end of the cylinder than in the head end and does not need the same amount of lead to secure full pressure at the beginning of the stroke. Both exhaust valves are given a 3/16-in. rise.

285. Q.—How does the valve gear operate on the Brown engine?

A.—The Brown engine is provided with two steam valves and two exhaust valves of the flat multiported



A285. STEAM AND EXHAUST VALVE GEAR OF BROWN ENGINE

type, each set being operated by a separate eccentric and all independently adjustable. Steam valves are placed on the side of the cylinder, and the exhaust valves at the bottom. To operate the steam valves, Fig. A285-1, the eccentric is connected to a lever, the lower arm of which is a steel lifting block which engages a latch. As the eccentric is drawn toward the crankshaft, the lifting block will be raised and carry with it the latch and the guide, causing the valve to open. This upward movement continues until the tail of the latch comes into contact with a trip lever, which causes the latch to release the block and allows the valve, with the stem and guide, to fall to their initial position, thus cutting off suddenly the admission of steam.

In operating the exhaust valves, Fig. A285-2, the eccentric operates a slide bar which is connected by a link and lever to an exhaust-valve stem. By this arrangement, in transferring the motion of the slide bar to the exhaust-valve stem, the greatest velocity of valve movement occurs at the points of opening and closing, and a long period exists between these points.

286. Q.—Why is it you cannot employ an unbalanced slide valve on an engine with a shaft governor?

A.—In the case of an unbalanced slide valve, the pressure of the live steam acting upon its outer surface results in a force due to the friction that must be overcome in moving the valve.

A balanced valve is employed in the shaft-governor type of engine because of the fact that with a high or medium speed engine, the use of an unbalanced valve would result in large valve friction which would make it difficult for the governor to control the movement of the valve.

287. Q.—On a riding cutoff engine, why does the rider have to have the same throw as the main eccentric?

A.—On a riding cutoff engine, it is necessary for the auxiliary or cutoff eccentric to have the same throw as the main eccentric, in order that it may have the same amount of motion, thus avoiding any possibility of the valve being reopened after cutoff has occurred. The auxiliary eccentric is so set that, when cutoff takes place, the cutoff valve has a motion opposite in direction to that of the main valve, and to secure a combined

motion of the two valves that will give the proper cutoff, one valve must have the same travel as the other.

288. Q.—With a Meyer valve, is the cutoff eccentric set ahead of or behind the main eccentric?

A.—The cutoff eccentric is set ahead of the main valve eccentric.

289. Q.—Are all four valves on a Corliss engine ever closed at the same time?

A.—In the single eccentric Corliss when the wrist plate is in its central position all four of the valves are closed by the amount of their lap. If, however, the exhaust valves are set with negative lap this would not be the case, because one valve would start to open before the other was entirely closed. In the double eccentric Corliss with long range cutoff all the valves are not closed at any one time as the admission valves may close as late as 7/8 of the stroke.

290. Q.—Why are two eccentrics used on a Corliss engine?

A.—In order to obtain a greater range of cutoff in a Corliss engine, separate steam and exhaust eccentrics are used.

291. Q.—What is the longest range of cutoff with a single eccentric Corliss engine?

A.—From the beginning up to one-half of the stroke.

292. Q.—What is it with two eccentrics?

A.—About seven-eighths, or eleven-twelfths of the stroke. Just before the steam valve on the opposite end picks up for lead.

293. Q.—In a 16 by 20-in. steam engine, cutoff takes place when the piston has moved through 12½ in. of its stroke. If the clearance space has a volume of 200 cu. in., what is the apparent cutoff, the real cutoff, and the ratio of expansion?

A.—Apparent cutoff is the ratio of the portion of the stroke completed at cutoff to the total length of stroke. The real cutoff is the ratio between the volume of steam in the cylinder and clearance space at cutoff and the volume, including clearance, at the end of the

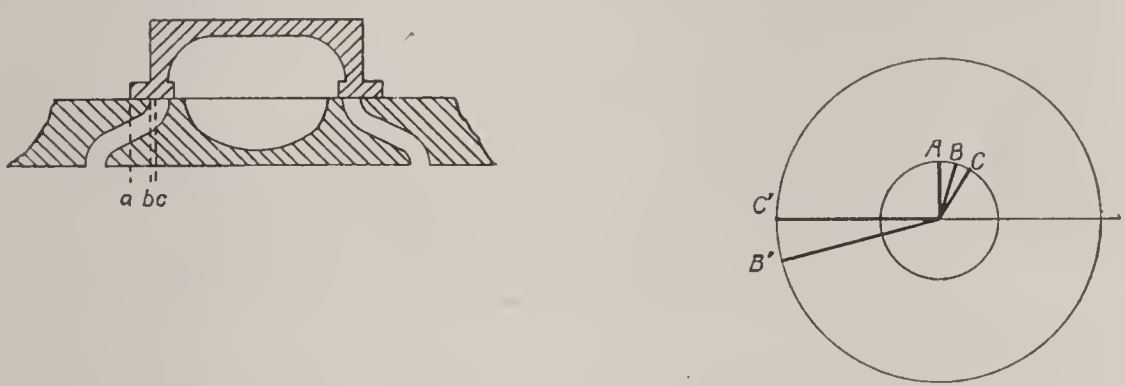
stroke. The ratio of expansion for a simple engine would be the reciprocal of the cutoff.

In the case stated, the apparent cutoff would be $12.5 \div 20 = 0.625$ or 62.5 per cent. Real cutoff would be found as follows: Volume of piston displacement = area of piston times stroke. Area of piston $= 3.1416 \times d^2 \div 4 = 3.1416 \times 16^2 \div 4 = 201$ sq. in. Stroke = 20 in. Volume = 2020 cu. in. Total volume, piston displacement plus clearance = 2020 + 200 = 2220 cu. in. Volume at cut off $= 201 \times 12.5 + 200 = 1462.5$. Cutoff $= 1462.5 \div 2200 = 0.66$ or 66 per cent.

The apparent expansion ratio for a simple engine with 62.5 per cent cutoff would be $1 \div 0.625$ or 1.54, and the real ratio of expansion $1 \div 0.66 = 1.51$.

294. Q.—What is the difference between angle of lap and angle of lead?

A.—The lap angle, meaning the steam lap, is the angle AB, Fig. A294, through which the eccentric must



A294. LAP AND LEAD ANGLES

be rotated from its central position A to cause the steam edge of the valve a to travel the distance of the lap ab. The lead angle BC is the angle between the position of the crank B′ when the valve begins to be opened, and its position C′ when the piston is at the beginning of the stroke.

295. Q.—Why are the valves of a single eccentric Corliss engine given (a) lap and (b) lead?

A.—(a) A Corliss engine valve is given lap in order to prevent steam from blowing through to the exhaust when the engine is carrying the hooks. In other words, should the engine become overloaded, the governor would fall and the valves would not trip and cut off the

steam before the end of the stroke, and if the valve was not provided with lap, steam would blow through the exhaust valve on part of the return stroke; but with lap, the valve is closed, even though the hooks have not tripped, before the piston starts on its return stroke.

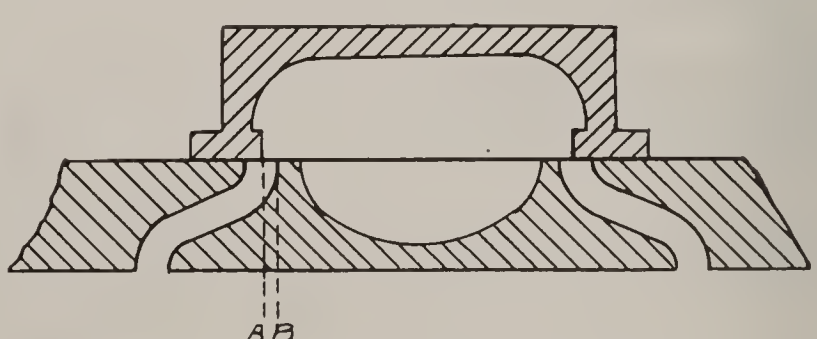
(b) Lead is given the valve so that by having the valve open a slight amount before the piston reaches the end of its stroke, the piston may receive the maximum pressure in reversing its direction. The wider opening through which the steam can flow during the early part of the stroke tends to reduce the wiredrawing action.

296. Q.—If you had a slide valve engine with too much inside (exhaust) lap, what would be the effect?

A.—The effect of this condition would be late release and early compression.

297. Q.—What is negative exhaust lap and why is it used?

A.—In cases where the valve has small steam lap or excessive travel, it is sometimes advisable to design the



A297. SHOWING NEGATIVE EXHAUST LAP

valve with negative exhaust lap, in order that the steam may be free to exhaust. Negative exhaust lap is the opening AB, Fig. A297, between the exhaust edge of the valve and the edge of the steam port when the valve is in its central position.

298. Q.—If lap is not changed, but the angle of advance is increased, what is the effect on the events of the stroke?

A.—Increasing the angle of advance will bring all events in the stroke earlier, but will have no effect on the amount of the port opening.

299. Q.—Has a high speed engine as much lead as a low speed?

A.—A high speed engine requires more lead than a low speed engine, with the same capacity, so that the clearance may fill quickly with steam as the piston starts forward.

300. Q.—What is the difference between changing the angle of advance and changing the throw of the eccentric?

A.—The throw of the eccentric is the distance from the center of the engine shaft to the center of the eccentric, and can only be changed by moving this center of the eccentric towards or away from the center of the shaft. This is done in many shifting eccentric governors.

The angle of advance is the angle by which the eccentric center is set ahead of its normal position for a D-valve, which had no lap or lead. This normal position would be, with a direct-acting valve motion, and with a valve taking steam on the outside ends, 90 deg. ahead of the crank (at A, Fig. A294, when the crank is at C′). With a reversing valve motion, or with a valve which takes steam on the inside, the normal position of the eccentric would be 90 deg. behind the crank. When lap and lead are added to a valve, it is turned ahead from its normal position enough to give it opening at the desired piston position before the dead center (AC, Fig. A294).

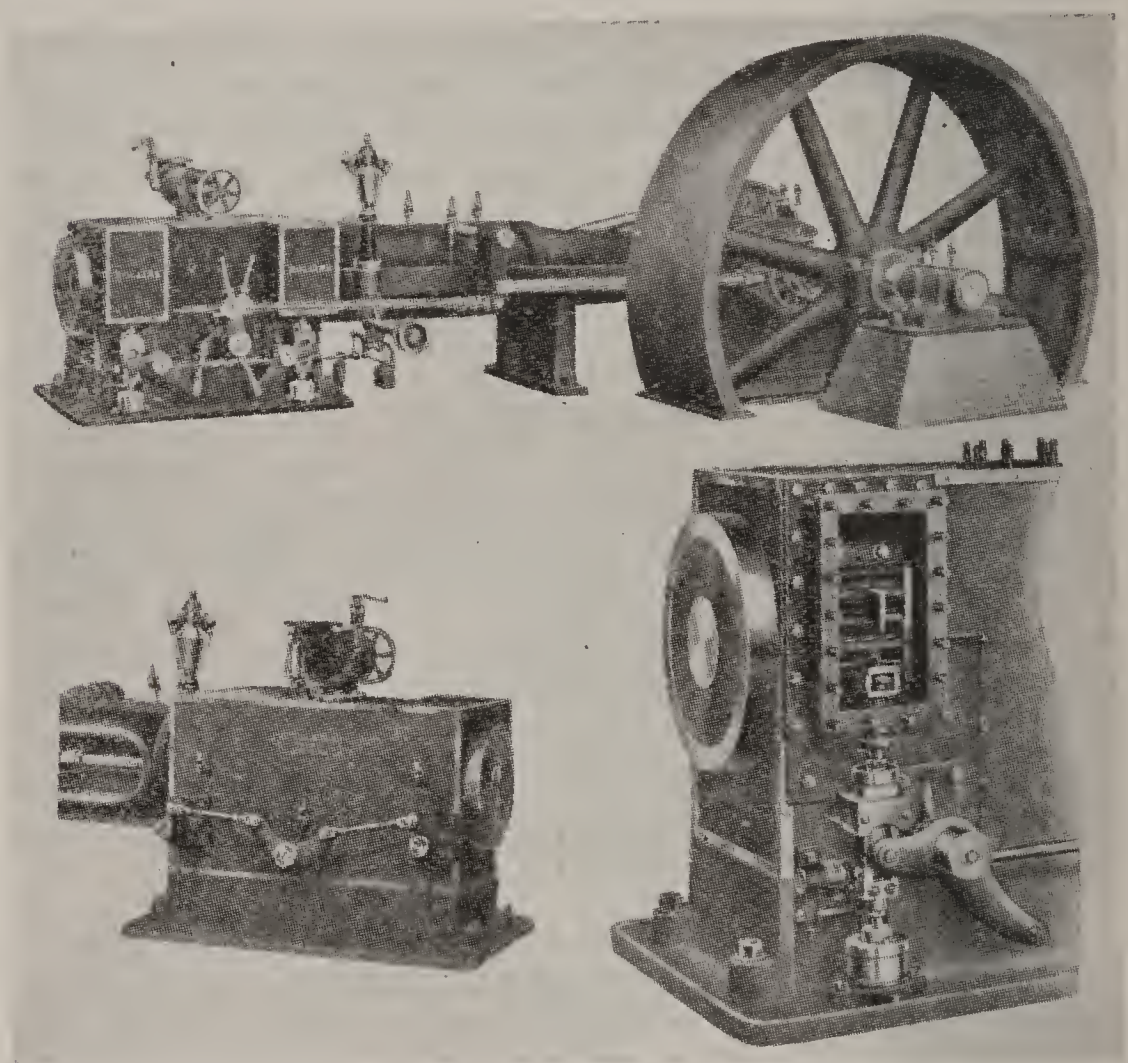
To change the angle of advance, we should have to set the eccentric so that it had a greater angle with its normal position. This may be done either by turning the eccentric around on the shaft, or it may come from swinging the eccentric about a pivot in such a way that it is moved forward in the direction of rotation with reference to the crank. As a matter of fact, most shifting eccentrics change both the throw and the angle of advance as the governor operates.

301. Q.—How can the lap of the valve on a Rollins engine be changed?

A.—The lap of the Rollins engine valve would be changed by changing the length of the link, or rod, between the valve proper and the releasing mechanism, or as you might term it, changing the length of the valve stem.

302. Q.—What is the earliest cutoff that can be given a D slide valve?

A.—The easiest way to study valve action is by the Bilgram valve diagram, Fig. A302-1. In this, AB represents the stroke of the engine, and A′ B′ the diameter of the crank circle. Oo is the throw of the eccentric, ed the lead, oL the steam lap, ol exhaust lap and oOB′

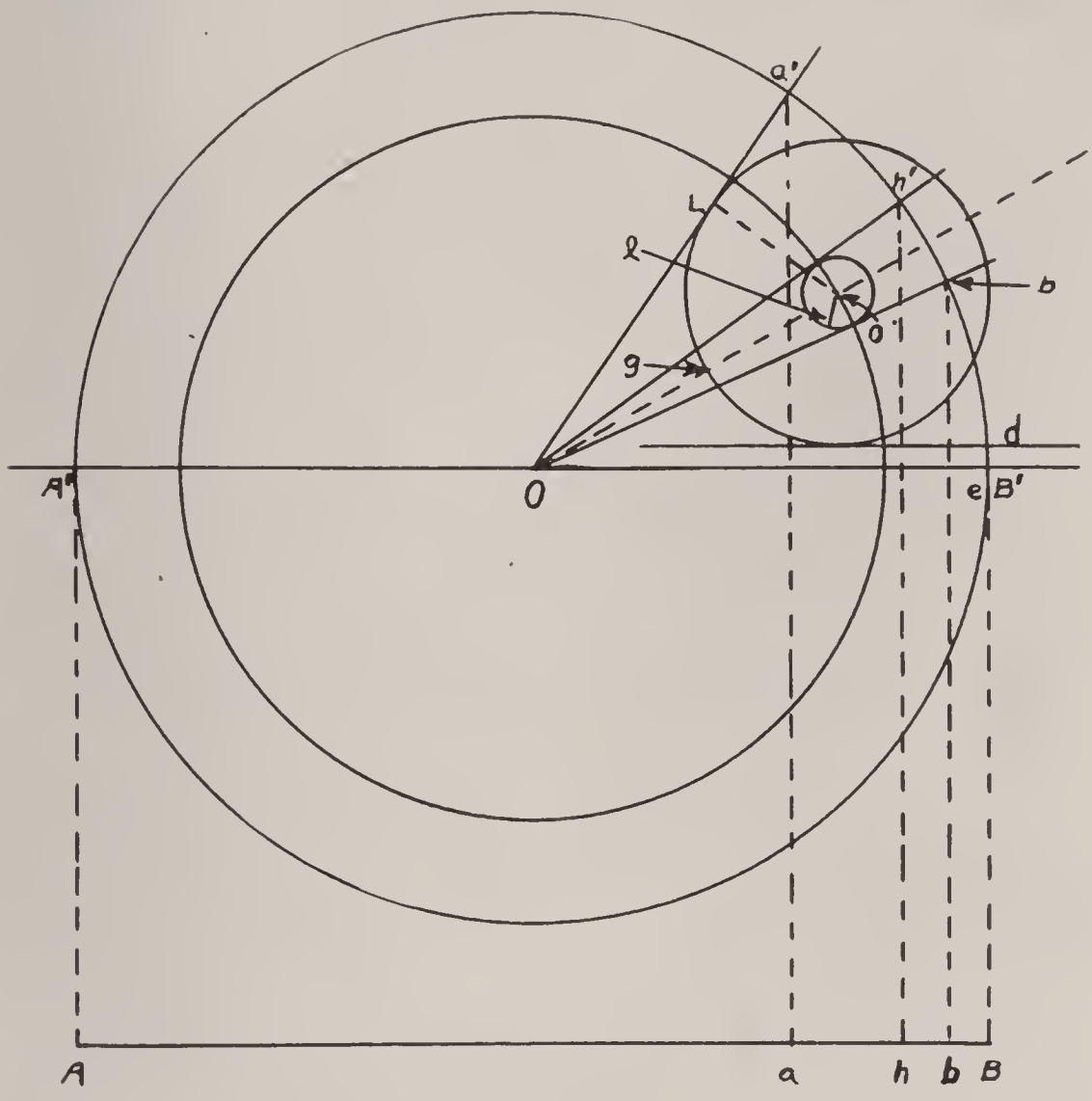


A301. DIAGRAM OF ROLLINS VALVE AND GEAR

the angle of advance. For Fig. A302-1 lead is 1/16 in., eccentric throw 1⅛ in., steam lap, ½ in., exhaust lap ⅛ in. negative. Steam port opening at extreme travel is shown by Og, ⅝ in., cutoff position head end for the crank is at Oa′, and piston positions at a for cutoff, at h for release head end and at b for compression, crank end.

To get cutoff at ½ stroke, as in Fig. A302-2, and keep port opening ⅝ in. eccentric throw has to increase to 2 in., Oo, steam lap to 1⅜ in. With exhaust lap zero, this brings release and compression at b and h 85 per cent of the stroke. As cutoff is brought earlier, travel

and lap increase, thus increasing the size of the valve and the power used up in friction. Also it is difficult to get proper release and compression. For these reasons, cutoff is not usual with a D slide valve at earlier than 5/8 stroke, although by allowing negative lead, valve opening after the crank has reached dead center, cutoff can be carried down to 3/8 stroke, as shown in Fig.



A302-1. BILGRAM DIAGRAM FOR D VALVE

A302-3, but with extreme results as to valve travel and lap, the former, twice Oo, being 7.5 in. and the latter, og, 3 1/8 in.

303. Q.—WHY ARE poppet valves used on some of the largest engines?

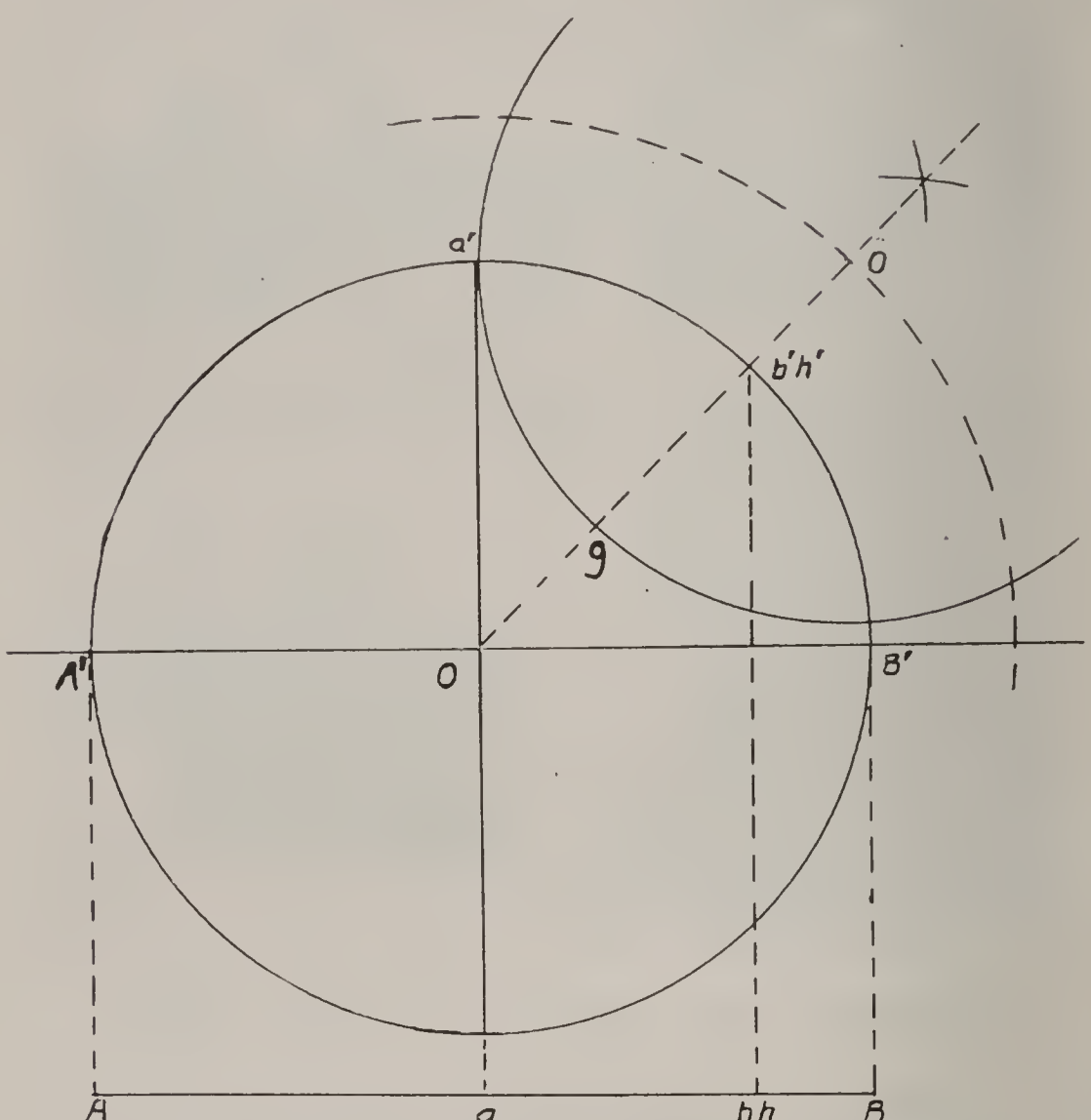
A.—One of the principal reasons for using poppet valves on large engines is, that superheated steam may be employed to better advantage than with slide valves or rotary valves. Poppet valves may be kept tighter,

as the seats and valves are more easily machined and ground.

Valve Setting

304. Q.—How could the cutoff of a slide valve engine be shortened?

A.—Cutoff on a D valve engine can be shortened by turning the eccentric ahead, but this will ordinarily



A302-2. Diagram for cut-off at 1/2 stroke

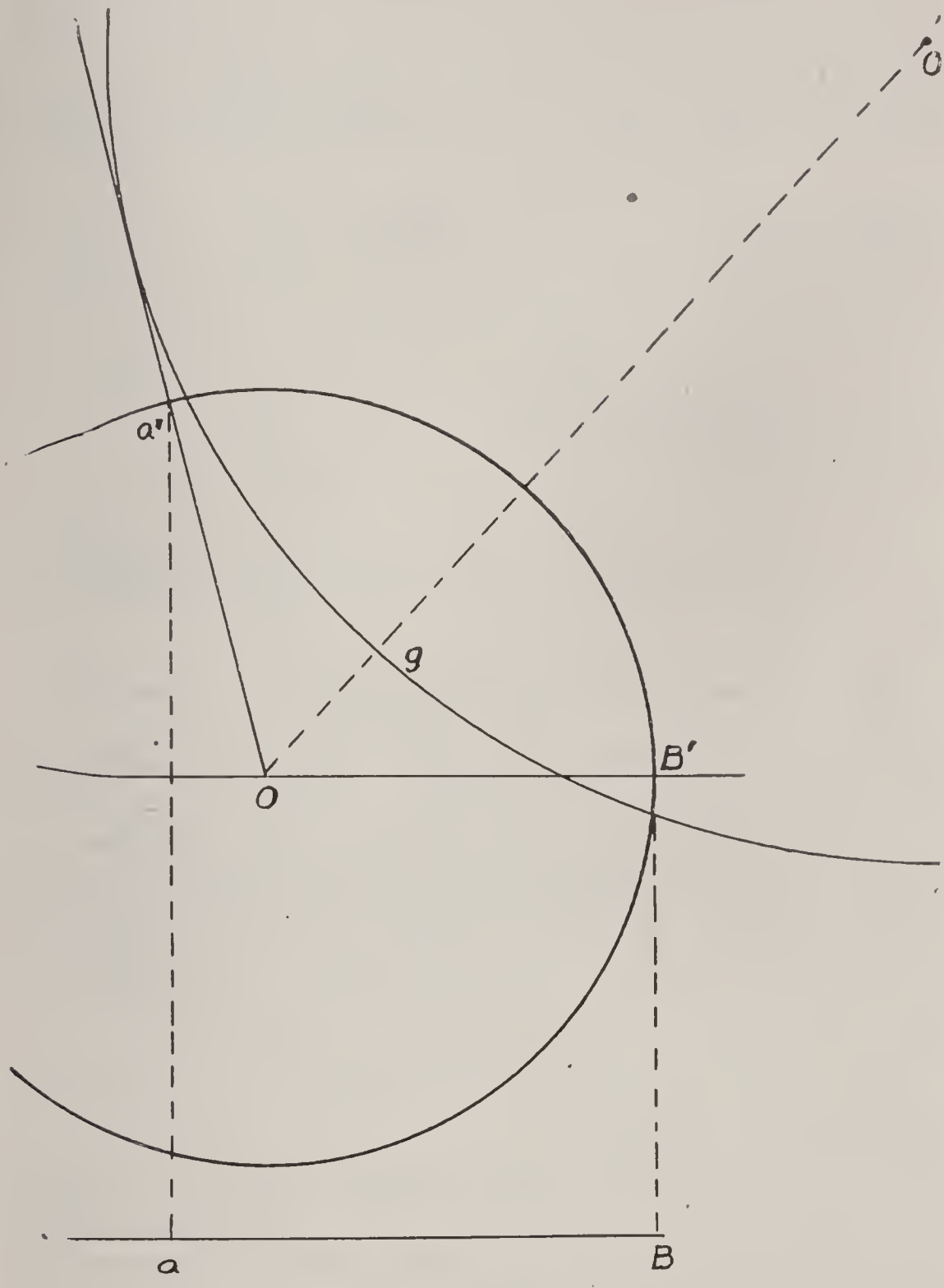
make trouble with other events in the stroke, so that it is not desirable to do it. It could, also, be accomplished by shortening the travel of the valve, but this again would be likely to make the port opening too small and produce wire-drawing.

305. Q.—If a slide valve has 3/4 in. lap with no lead, how could it be given 1/32 in. lead?

A.—A slide valve with no lead can be given lead by

turning the eccentric ahead with the piston on dead center, until the valve shows 1/32 in. opening.

306. Q.—How can a slide valve be given a travel greater than twice the eccentric throw?



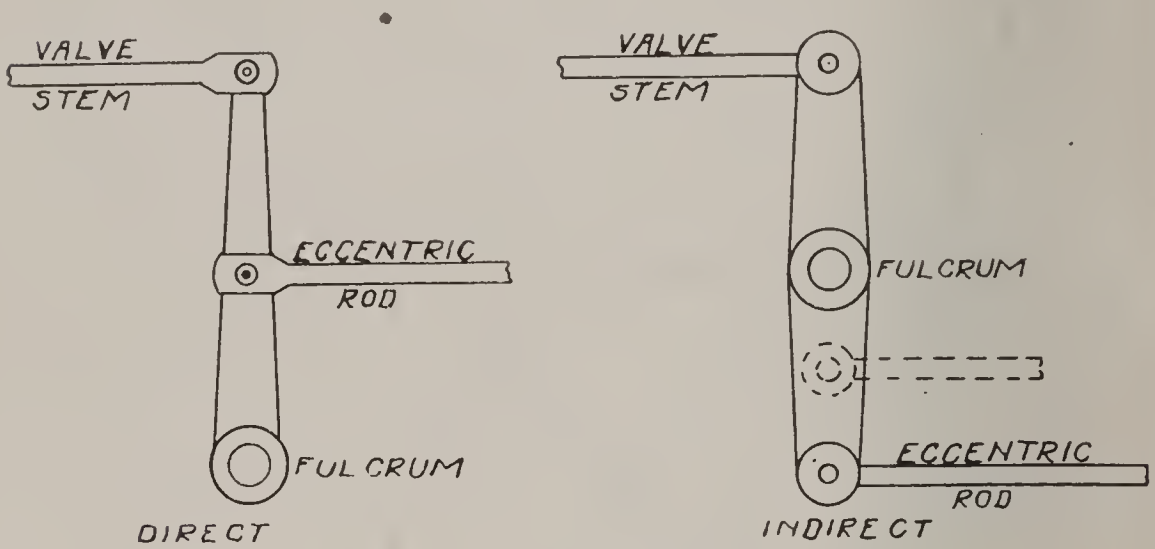
A302-3. DIAGRAM FOR CUT-OFF, 3/8 STROKE

A.—The travel of a slide valve may be increased by means of a carrier arm to which both the eccentric rod and valve stem are attached with the eccentric rod nearest the fulcrum, as in Fig. A306. If the valve

motion is "indirect" the arm of the rock-shaft to which the eccentric rod is attached may be shortened.

307. Q.—How is a plain slide valve set?

A.—Loosen the eccentric on the shaft and set it at mid position. The valve should travel equally on both ends, or in other words, should be at mid position when the eccentric is at mid position. This equality of travel on both sides of the center line applies also to the intermediate rocker arm situated between the valve and ec-



A306. DIRECT AND INDIRECT ROCKER ARM VALVE GEAR

centric of an indirect valve gear. Place the engine on the dead center, usually the one nearest the cylinder, rotate the eccentric on the shaft in the direction in which it is intended to run until you have the proper amount of lead. Make it fast and, if possible, place the engine on other center and see whether you have the same lead. If not, equalize lead by lengthening or shortening the valve stem by half the difference.

308. Q.—How would you set the piston valve on a Westinghouse engine?

A.—In the smaller sizes of the Westinghouse engines the shaft governor is attached to the main shaft in the crankcase between the two cranks and of course cannot be changed from its position as located by the maker, so that all that can be done is to equalize the valve travel between the two cylinders the same as would be done in equalizing the valve travel of a piston valve on any high speed engine. If possible withdraw the valve and with a scale transfer the ports to the side of the steam chest

with a sharp scriber. Make a record of the valve itself with its ports, and then by careful work, the valve action can be recorded as accurately as a slide valve with the chest cover off.

309. Q.—What is the procedure in setting Corliss engine valves?

A.—Although directions for valve setting are numerous, the following have some methods not commonly given and which are worth remembering:

First, remove the steam and exhaust bonnets and upon the end of each valve will be found a mark corresponding to the opening edge of the port. The direction of motion for opening the valve can be determined by working the wristplate by hand.

Second, examine the wristplate hub and bracket. Upon the bracket will be found a mark corresponding to the central position of the wristplate and two others locating the extreme travel of the wristplate in either direction of motion.

Third, see that the rocker arm travels an equal distance each side of the vertical position. To do this, loosen the eccentric and rotate it slowly about the shaft. Then with a plumb bob and scale measure the distance the rocker arm travels each side of its central position, adjust the length of the eccentric rod to bring the distances equal.

Fourth, rotate eccentric about the shaft and see if the wristplate travels equal distance each side of its central position, as shown by the travel marks. If it does not, adjust the length of the hook rod to bring them equal.

Fifth, block the wristplate in its central position, then pull up the dashpot rods until the hook pins engage with the steam hooks. Adjust the length of the steam and exhaust radius rods until the valves have the proper amount of lap as determined by the size and speed of engine.

Sixth ,throw the wristplate with the starting bar first to one, then to the other extreme position, and adjust carefully the length of the dashpot rods so that the steam hooks will latch with the hook pins when the wristplate is at its extreme travel, allowing 1/32 in. for clearance.

Seventh, place the engine on dead center and see that the hook rod is fastened to the wristplate. The

eccentric should then be turned in the direction the engine is to run until the proper amount of lead is secured, fasten the eccentric to the shaft. Place the engine in the opposite dead center, and measure the lead. If it is not the same, make it the same by adjusting the length of the steam radius rods.

Eighth, block the governor up half way in the slot and fasten the reach rod levers at right angles to a line drawn half way between the two rods. Adjust the length of the reach rods until the knockoff arm stands vertical; with the governor in this position, turn the engine over until the cutoff occurs, and measure carefully the distance the crosshead has traveled; place the engine so that it will be the same on the other side, and adjust the length of the reach rod until the cutoff will just occur. This insures equal cutoff at each end of the cylinder for these positions. See that the governor gag pot is properly adjusted, and that the pot is filled with light oil.

Ninth, drop the governor to its lowest position and see that safety blocks on the knockoff cams are adjusted so as to prevent the steam hooks from latching with the hook pins.

Tenth, tram your engine with a fine prick punch so as to be able to adjust it quickly in case of accident or in case it has been tampered with.

310. Q.—How are the valves set on a double eccentric Corliss engine?

A.—The work of centralizing the positions of the various parts, equalizing the movements and setting and adjusting the gear is practically the same as with the single eccentric engine. Set the wristplates central, and adjust the valve rods, giving the steam valves negative lap, usually a little less than half the port opening.

The first step then is to set the exhaust eccentric (as it is placed next to the bearing). Turn the engine until the piston is in the desired position, so as to obtain a compression of about 5 per cent of the stroke, then turn the exhaust eccentric loose on the shaft, in the direction the engine is to run, until the exhaust valves are line and line, then secure the eccentric.

The next step is to set the steam eccentric; place the crank on either one of its dead centers, then turn the steam eccentric loose on the shaft until the steam valve

at the end where the piston stands has the required opening or lead, which varies from 1/32 in. to ⅛ in., according to the size of the engine and speed it is to run.

311. Q.—How would you set the valves on a triple expansion Corliss engine?

A.—The method of setting the valves on a triple expansion engine of Corliss type is precisely the same as that employed for a simple engine, except the adjustment of cut-offs in the various cylinders, which is largely a matter of "cut and try" until best results are obtained. This is usually considered to be an equal load on each of the cylinders.

An increase of cut-off of the high pressure cylinder while the initial steam pressure and the cut-offs of the intermediate and low pressure cylinders remain as before, results in an increase of the pressure in the first and second receivers, and increase of the mean effective pressure in all three cylinders, and consequently an increase in the horsepower of the engine.

Conversely, an earlier high pressure cutoff, without change of the other cutoffs and initial steam pressure, means a decrease in receiver pressures, mean effective pressures and horsepower.

A lowering of the initial steam pressure in the high pressure cylinder that is not accompanied by any change of cutoffs, results in lower receiver pressures, lower mean effective pressures, and lower horsepower.

Conversely, if under the same conditions the initial steam pressure is raised, the receiver pressures, the mean effective pressures and the horsepower will be increased.

Other conditions remaining the same, if the intermediate cylinder cutoff is made later, the pressure in the receiver of the intermediate cylinder is lessened, and the drop from the high pressure increased, whereby the power developed in the high pressure cylinder, owing to increased mean effective pressure, is increased. At the same time the mean effective pressures in the intermediate and low pressure cylinders and hence in the horsepower there developed will be decreased. The converse of this is also true.

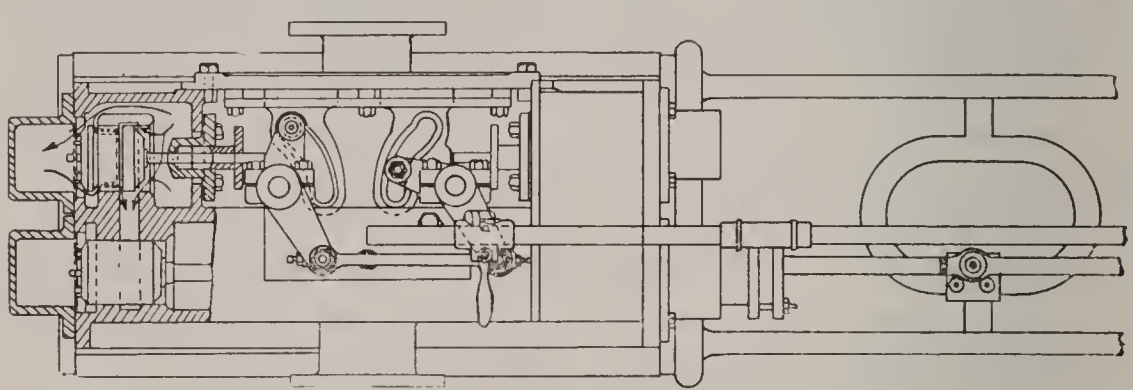
Again, suppose that other conditions remaining the same the lower pressure cutoff is made later. Then the pressure in the second receiver will be lower, the mean effective pressure in the low pressure cylinder and hence

the horespower there devoloped will decrease. At the same time the mean effective pressure in the intermediate cylinder, hence the horsepower there developed will be increased, but no change will take place in the high pressure cylinder.

If the low pressure cutoff is made earlier, under the same conditions the horsepower developed in the low pressure cylinder will be increased—that of the intermediate cylinder will be decreased, and that of the high pressure cylinder will remain the same.

312. Q.—Tell how to set the valves on a Fitchburg engine.

A.—To set the valves of the Fitchburg engine, first adjust all rods and connections so that the several parts will travel equally on both sides of the center in a com-



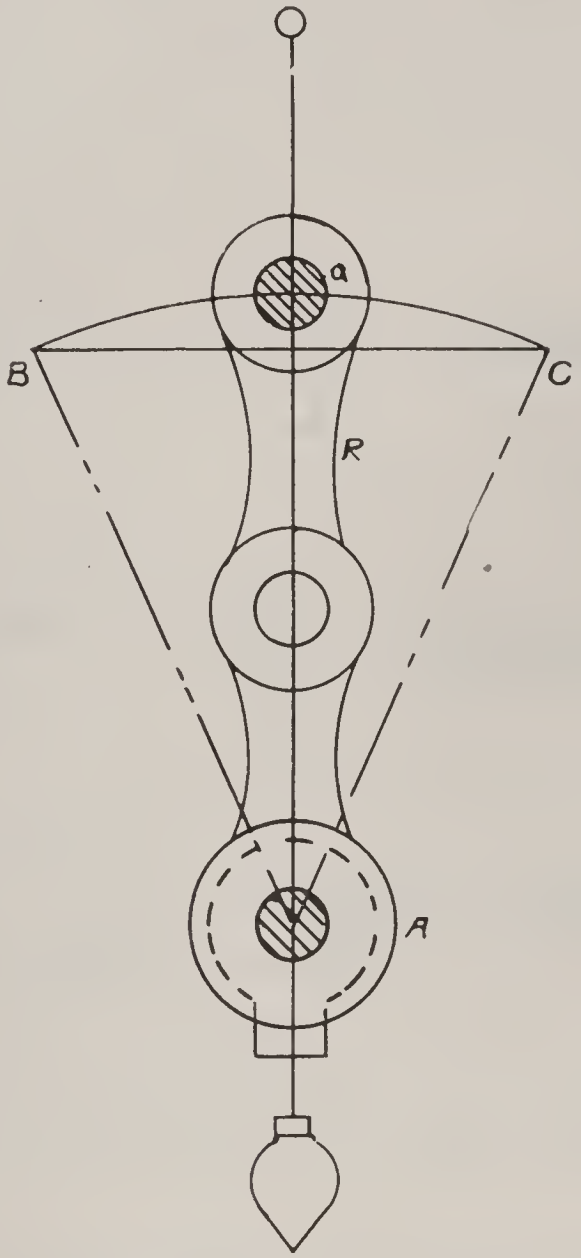
A312. FITCHBURG VALVE GEAR

plete revolution of the engine. As the engine has two eccentrics, one for the steam valves and one for the exhaust valves, they should be set separately. Ignoring for the present the exhaust valves, place the latch of the steam valves in the center of the spiral slot and clamp the hook by its lever, adjusting the movement of the wrist cranks by the right and left connections as provided. Set the engine exactly on the head center and move the small rod attached to the head valve in or out of its cam until the port is opened the proper lead (which is usually about 1/16 in.) and tighten the set screw in the neck of the rod. Roll the engine on the other center and proceed as before.

To set the exhaust valves adjust for equal travel as usual, and set the eccentric on the shaft so that the exhaust valve at any given end will be line and line and closing when the piston is about one-eighth of the stroke from the end. It is best to "spot" all setscrews to prevent slipping.

313. Q.—Give directions for setting the valves on a tandem-compound Buckeye engine.

A.—The operation of setting the valves of a Buckeye engine is quite simple. First see that pin a at the upper end of rocker R vibrates an equal distance each side of its central position. This may be done by plumbing up



A313-1. ROCKER ARM OF BUCKEYE ENGINE

from pivot A, Fig. A313-1, and measuring the distances aB and aC, which represent the horizontal travel of the pin. If these are not equal, adjust the fixed eccentric rod F, Fig. A313-4, until they are equal. For future reference, marks indicating the corresponding positions may be made on the hub or other convenient point, if they are not already there.

Next place the engine on the head-end center, being careful to take up all lost motion in the direction of

rotation. Unhook the fixed eccentric and place the rocker arm R in its central position. The main valve should then be in its central position on the valve seat. Now roll fixed eccentric around the shaft, to such a position behind the crank that the next valve movement will be away from the crank, or contrary to the forward stroke of the piston, as shown in Fig. A313-2; couple the eccentric rod to pin a of the rocker. Then leaving the engine on the dead center, roll eccentric F further around the shaft toward the crank until the outer edge

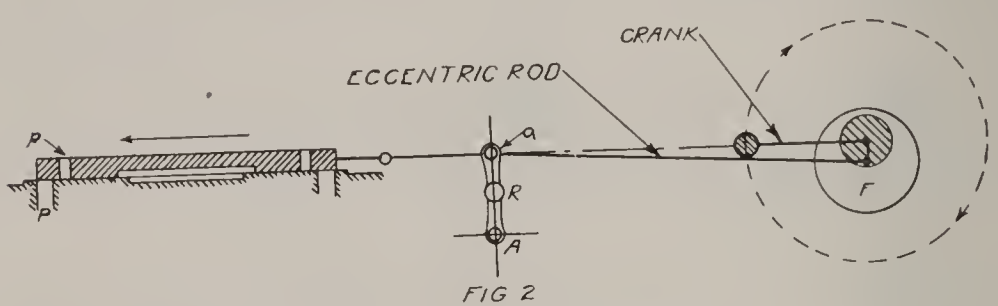


FIG 2

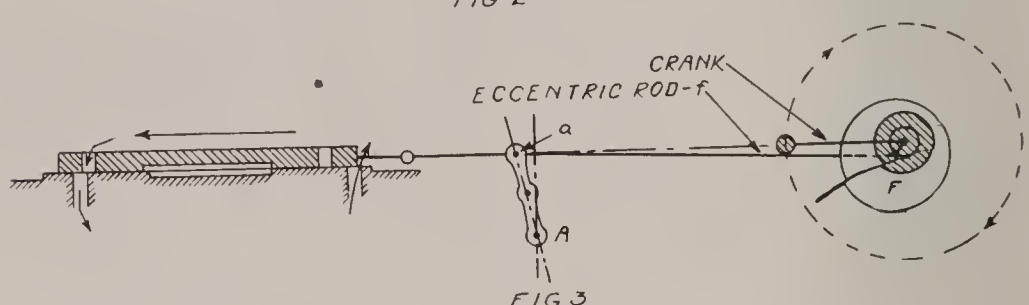


FIG 3

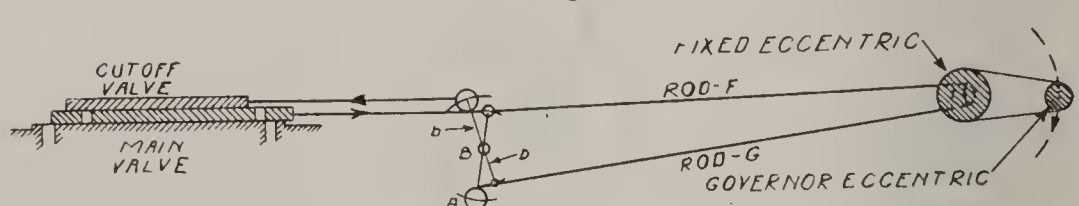


FIG 4

A313—2. RELATION OF VALVE, ROCKER ARM, ECCENTRIC AND CRANK, MID-POSITION; 3. POSITION FOR LEAD; 4. MAIN AND CUTOFF VALVE GEAR

of valve port p overlaps the inner edge of cylinder port P by the amount of the lead, or until the port edges are line and line, if there is no lead. The valve ports, crank and eccentric will now occupy a position about as Fig. A313-3. Lock the eccentric F in place and roll the engine around to the crank center; if the work has been accurately done, the valve will show proper lead on the crank end. Slight adjustments may be made by the valve stem. This completes the setting of the main valve. If the eccentric is but slightly out of place, it will be necessary only to adjust the eccentric to show proper lead.

For the cutoff valve, first see that it makes the proper length of stroke on the main valve, adjusting eccentric rod G, Fig. A313-4, until it does. Then with the engine on the crank-end center, allow the governor to come to rest on its inner stops. This will revolve governor eccentric backward on the shaft until its center about coincides with the crank. The exact position of governor eccentric when the governor is at rest will be fixed by the design of the engine; for condensing engines it may be a little ahead of the crank, while for heavily loaded engines not subject to great fluctuations of load, it may be a little behind the crank.

Under ordinary conditions the crank and the governor eccentric should pass the dead center at the same time, in the same direction.

Then, with the engine on the crank-end center and the governor eccentric coinciding with the crank, the cutoff valve should be in its extreme left-hand or head-end position, since the rockers b, Fig. A313-4, reverse the motion of the eccentric. If the valve does not occupy this position, adjust the valve stem until it does. With crank and eccentric still coincident, turn the engine over to the head-end center and see that the cutoff valve is in its extreme crank-end position. The point of cutoff is then determined by the angular advance of the eccentric, which in turn depends upon the adjustment of the governor. Roughly speaking, for one-quarter cutoff the governor should occupy such a position as to bring the governor eccentric about 90 deg. ahead of the fixed eccentric.

314. Q.—How are the valves of a Greene-Wheelock engine set?

A.—The following instructions are from the builders of these engines, and if adhered to will give proper setting of the valves.

For reference and a means of checking off the action of the valves, it is stated that "A-size" valves have 5/32-in. lap, with ¾-in. travel, and are generally used on cylinders up to and including 16 in. in diameter; "B-size" valves have 3/16-in. lap, with 1⅛ in. travel, and are generally used on cylinders from 18 to 26 in. in diameter, inclusive; "C-size" valves have ¼-in. lap, with 1⅜ in. travel, and are used on cylinders from 28 in. in diameter upward.

When starting to adjust the valves, first have all eccentrics loose on the cylinder shaft, and, second, determine the direction that shaft is to run, and always rotate the eccentrics in the same direction, whether loose on the shaft, or when the shaft and eccentrics turn together.

On the edge of the pusher crank, Fig. A314-1, a line is made in the shop, and on the side of the plug head, next to the pusher crank, a corresponding line is made (where the arrow points). When the line on the pusher crank corresponds exactly with the line on the side

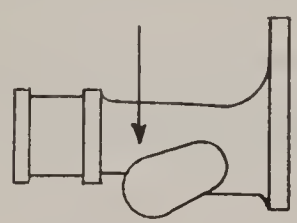


FIG. 1

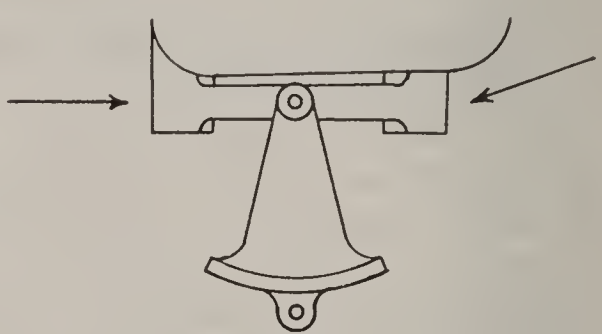


FIG 2

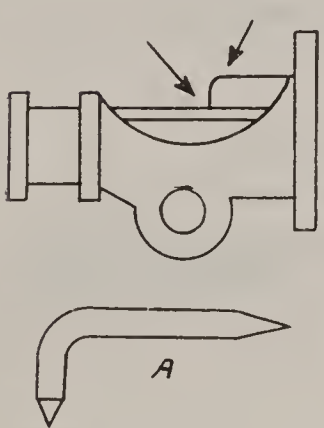


FIG. 3

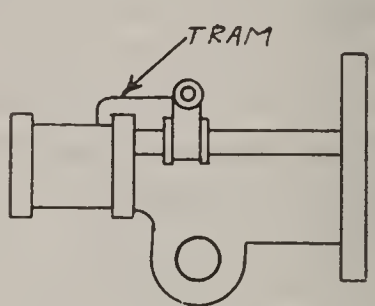


FIG. 4

A314. VALVE GEAR DETAILS OF GREENE-WHEELOCK ENGINE

of the plug head, the pusher plate is vertical. This is its most backward position.

Adjust the eccentric rod for this valve to such a length that in turning the eccentric around on the shaft the line on the edge of the pusher crank comes back to correspond exactly with the line on the plug head at each revolution. Then, by shimming, adjust the bridge-supporting trip cam (shim where the arrows in Fig. A314-2 denote), so that the steam valve will travel ¾ in. on "A" size, 1⅛ in. on "B" size, and 1⅜ in. on "C" size, but bear in mind that the valve must trip at the end of its travel and the bridge must not be so

low that the valve will carry the full stroke without tripping. The roller of the lifter must be in position for full travel.

On the steam-valve stem four scratch lines are made. These lines represent the valve on its lap, the valve just opening, the valve wide open, and the valve pushed in until it strikes the plug. With each valve gear a steel-wire tram (Fig. A314-3) is sent. Just above the valve stem on the plug-head casting a prick-punch mark will be found. Loosen the inlet steam head on the stem, then shove the valve back until it strikes the plug. If the valve is set correctly, the tram with one end in the mark on the plug-head casting, as shown in Fig. A314-3, should with the other end meet the first scratch line on the valve stem (nearest the outside end of stem). If the point of the tram does not coincide with this line, the valve stem should be screwed on or out until it does. The valve should then be let back so that the dasher strikes the head, and the inlet stem head be brought back against the pusher plate when the pusher plate is vertical, leaving 1/64 in. clearance between the pusher and latch plates. It will then be found that the point of the tram will correspond with the fourth mark on the stem, with the valve closed.

When the valve is moved forward so that the tram point corresponds with the third line on the stem, the valve is just closing or opening, and when moved farther so that it corresponds with the second line, the valve is wide open. The travel of the valve should be between the second, third and fourth points spoken of, and it should trip just as the tram point corresponds with the second line from the outside end. Then, with the piston on dead center, the eccentric should be revolved on the shaft to bring the steam valve 1/32 in. open on the crank end and 3/64 in. on the head end. The eccentric should then be clamped to the shaft, and the valve is set.

On the outside of the plug head, where the arrow points in Fig. A314-4, are four prick-punch marks. On the outside of the outlet stem head where the tram rests is another prick-punch mark. This is for one point of tram.

The eccentric rod should next be disconnected from the eccentric. Shove the valve back as far as it will go. With the valve in this position, the outside end of the

tram should fall into the fourth mark on outside of the plug head nearest the cylinder. If it does not, loosen up the nut holding the outlet stem head, and screw the stem in or out sufficiently to make the tram come up into the fourth mark. Then tighten up the nut holding the outlet stem head, connect the eccentric rod to the eccentric, lengthen or shorten this eccentric rod so that the travel of the valve due to one revolution of the eccentric will move the tram from the first to the third prick-punch mark, and no farther.

The eccentric should then be set so that when the piston is about 5 in. from the end of the return stroke, the exhaust valve should have just closed, and the tram point would fall into the second mark on the plug head.

Bearings

315. Q.—What should be the length of the main bearing of an engine? What should be the thickness of babbit in bearing?

A.—There is a wide difference of opinion concerning the length of main bearing. Thurston's formula for determining length of a shaft journal is:

$$l = \frac{P\,V}{60{,}000\,d} \text{ or } l = 0.000{,}004{,}363\,P\,R$$

and Rankine's formula is:

$$l = \frac{P\,(V + 20)}{44{,}800\,d}$$

Where l = length of bearing
P = mean total pressure, in pounds on the projected area of the bearing
V = velocity of the rubbing surfaces in feet per minute
d = diameter of the shaft, in inches
R = revolutions per minute.

Unwin gives the formula: $l = 0.4$ hp. $+ r$, in which hp. is the horsepower transmitted to the crankpin and r is the crank radius, in inches. An allowance is made of from 400 to 600 lb. per square inch of projected area.

There is no formula available for determining the thickness of the babbit. The engine designer probably has a rule based upon his own judgment and the characteristics of the engine.

316. Q.—When would you take up on the quarter boxes on the main bearing of an engine? Would you do it with the engine running or stopped?

A.—When there is evidence of lost motion. The engine should be running and the adjustment made gradually. Care should be taken that both quarter boxes are drawn up exactly the same.

317. Q.—How can you tell when a bearing is taken up enough?

A.—When a bearing is loose, there is a knocking noise and it is liable to get hot; in taking up on it, the knocking will cease when it is taken up enough, but if taken up too much, it will become hot again, so that if there is no knocking, and if the bearing does not become hot, it is safe to assume you have taken up enough.

318. Q.—How would you take up on the quarter boxes of the main bearings of an engine in operation?

A.—Main shaft bearings with quarter boxes have the adjustment in one of two ways; either a wedge adjustment, in which the wedge forcing the quarter boxes against the journal, is controlled by a nut and lock nut on top of the box, or by adjusting screws passing through the sides of the main bearing. In either case, the adjustment is made by turning up on the adjusting nut or screw a very little at a time until the play is eliminated. Great care should be taken in doing this, since there is always danger of getting the quarter boxes too tight, or if the adjustment is of any great amount, of throwing the shaft out of square with the cylinder.

319. Q.—For babbiting crankpin brasses, what is the proper way to go about it and the proper kind of babbit to use?

A.—We will assume that the engine is a small one and of side crank drive, and that the pin has not been worn out of round. Before dismantling the box, scribe around the collar, making a heavy mark on the brass box for future reference, as well be shown later.

In order to make a satisfactory job of any kind of babbiting, care must be taken to avoid "cold shots," by warming the pin and boxes, which insures rapid flow of the molten metal. Heat a piece of metal that will easily

slip over the pin and leave it there while you do the other necessary work. If no rings or castings are at hand, heat a solid chunk of iron, set it on an iron pedestal under the crankpin, and leave it there till ready to assemble the box.

While this heating up process is going on, remove the old babbit by heating over a forge on a thin plate, to conduct the molten metal to one side, or with one or two gasoline torches. If using an open fire to melt the babbit, take care not to burn the boxes.

Note whether the boxes are equipped with the necessary anchors to hold the new babbit. Also whether they are free from sand. If the anchors are not sufficient to hold the new babbit in place properly, drill holes, say with a ¼ or ⅜-in. drill at a slight angle in opposite directions in the boxes, and countersink them slightly to prevent breaking off, which sometimes happens if the corner is sharp.

Plug up any oilholes in the pin with a wooden plug driven in and the head covered with putty. This combination is easily removed with a gimlet afterward. Cover the pin with a light coat of black lead mixed with cup grease.

With the pin and boxes as warm as can be handled, assemble the bearing on the pin, using liners, say ⅛ in. thick and wide enough to extend in to the pin so that the two halves of the box will not be in one solid piece. The top liner can be arranged with holes to pour through or, if the strap has a hole for the oil cup, this can be used as a pour hole, and the metal afterwards drilled out.

Take care in this operation that the coating of black lead is not rubbed off, which may be avoided by blocking up the connecting rod with a suitable stick. Then revolve the shaft, or move the crosshead as required, till the collar on the pin coincides with the scribed mark on the brasses. Draw up snug on the key or wedges, and be sure that the brasses fit in the strap properly.

Divide up the clearance between the collar on the pin, and the crank disk. When all is well, pour the babbit, making it all in one pour, or if the job is too big, or the ladles too small, have the second ladle ready to pour as soon as the first is empty, so there will be no "break" in the babbit.

Babbit is one of the many things in this world that

few people are acquainted with, although in common use. Speeds and pressures are the principal factors entering into the use of babbit and the manner of mixing the several ingredients also has much to do with the lasting qualities.

If the engine is a center crank, or the bearing large, it is best to babbit it around a mandrel slightly smaller than the pin, and after pening it with a ball-pene hammer, assemble the two halves in the same relative position they have on the pin, and bore them to size.

If it is a strap end, wedge the boxes in it and clamp on the face plate of mill or lathe. If a solid end rod, use strap iron and bolts to hold boxes while boring.

After the boxes are babbited and bored, scrape to a fit on the pin, and relieve each half about an inch or more from the joint to prevent binding of the pin in case of slight heating. Cut generous oil grooves in both boxes, in such a manner that the oil will be continually worked toward the middle of the pin in preference to toward the ends, but do not have the grooves too wide, as this cuts down the bearing surface.

If you have time, after assembling the bearing, key up snug and run the engine slowly, watching carefully, and as soon as it begins to warm up, stop and scrape off the high spots. If necessary and convenient, repeat, and you will have a fine bearing.

Eccentrics

320. Q.—If you cut ¼' in. off all around the eccentric what changes would you have to make to operate the engine?

A.—If you turn ¼' in. off the eccentric all around, no changes will be required on the engine except to put on a smaller eccentric strap and lengthen the eccentric rod. The throw of the eccentric will not be changed.

321. Q.—If the eccentric was lost and you wanted to get a new one, how would you get the size?

A.—Find the valve travel by examining the steam chest, and determine how far the valve has been running back and forth. Then the eccentricity of your eccentric is equal to half the valve travel.

322. Q.—How should one proceed in rebabbiting an eccentric strap?

A.—To babbit an eccentric strap, first, make all necessary arrangements about shims between the straps before removing the old babbit, as this will help to keep the strap in the original position. See that at least one shim on each side extends down to the eccentric so as to act as a parting strip after pouring.

Prepare several small strips of wood of such strength as to brace the eccentric in the proper position and especially to support it from below. To prevent the babbit from running out the side of the strap make a set of "circles," out of some solid wood, free from knots.

These "circles" should be about 2 in. smaller in internal diameter, and the same in external diameter, as the diameter of the eccentric. These can be split and clamped against the eccentric and strap so as to cover the joint well, cutting away to pass any obstructions.

Have at hand, ready for instant use, a number of clamps fitted and arranged for use. Arrange for pouring the babbit by either making an opening at one side of the circles near the top, or else drilling a hole in each strap at the highest point for the purpose. Having arranged all the details, remove the straps and melt out the old babbit. Cover the eccentric with a coating of white lead, oil, or graphite and grease, to prevent "freezing" of the babbit to the eccentric.

While the straps are quite warm, clean them thoroughly and quickly assemble them again in position, bracing them in position same as before. Have the babbit heating in the meanwhile. Clamp on the wooden circles firmly and cover all places where there is the least possibility for the molten metal to get out, with putty, clay, or asbestos mixed with cylinder oil to the consistency of putty.

On small eccentrics arrange so that the metal will run into the two sides at the same time, but on larger eccentrics, have two men to pour and babbit both sides at the same time. See that you have enough metal on hand to finish the job at one pour. Allow the job to cool gradually in place and when cold enough to handle, remove all braces and other fixtures and take off the straps, when they can be cleaned up and oil grooves cut where necessary.

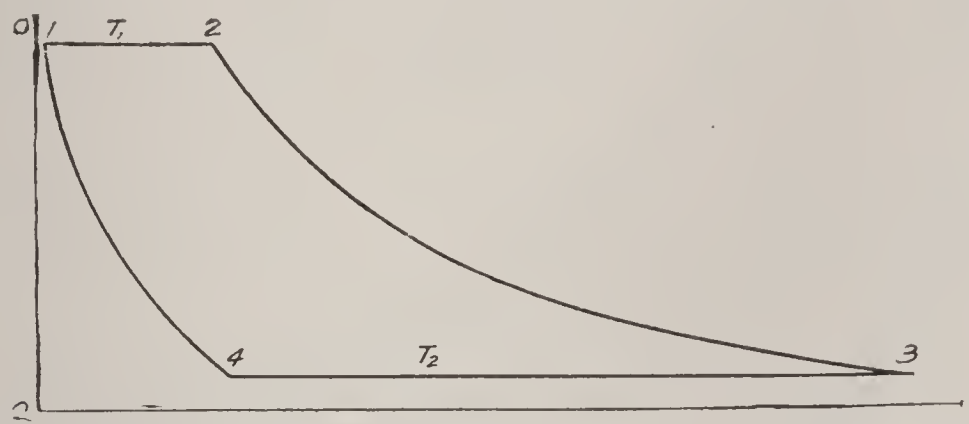
Then reassemble the straps on the eccentric adding

the necessary shims so that the straps will work freely. Note whether the babbiting has disarranged the valve setting. This can be done by tramming on a rod or stem from a convenient stationary part before starting and after finishing the job. If everything is all right, put on the oiling or other devices, and you are ready to run.

Indicator Cards

323. Q.—What is meant by the Carnot cycle?

A.—When a substance is caused to undergo a series of processes, for any purpose, and is finally brought back to the thermodynamic state it had initially, the substance is said to perform a cycle. In case the series



A323. CARNOT CYCLE FOR DRY AND SATURATED STEAM.

of processes which the working fluid undergoes in any cycle are all reversible processes, the cycle is said to be reversible.

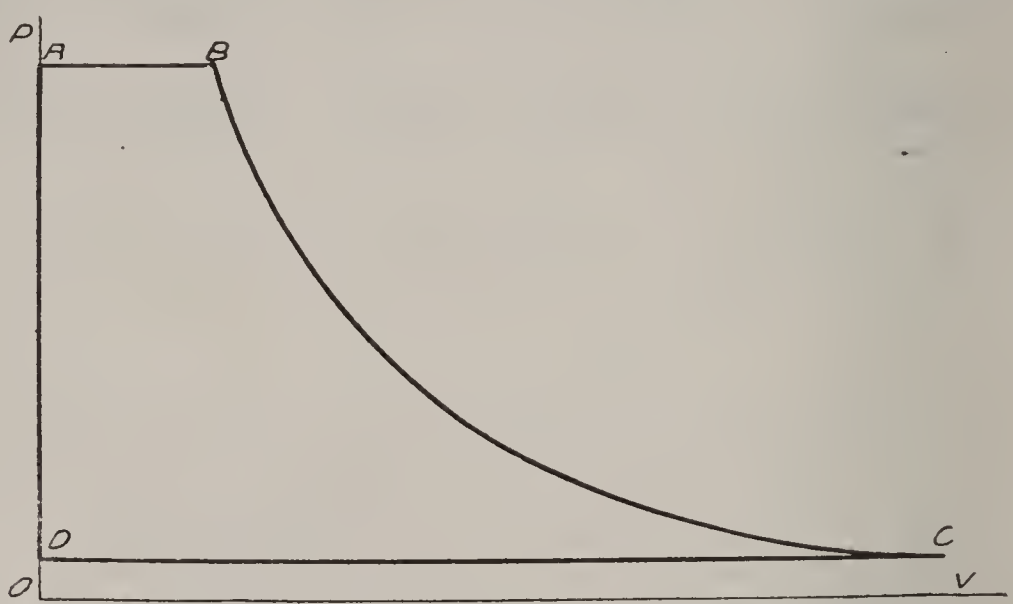
The simplest form of reversible cycle is that devised by a French engineer by the name of Carnot, and known, therefore, as the Carnot cycle. In this, the working substance undergoes the four reversible processes indicated in diagram Fig. A323. First, the substance is caused to expand isothermally at some temperature, T_1, from state 1 to state 2. It is then caused to expand adiabatically from state 2 to state 3, thereby attaining some lower temperature, T_2. It is then compressed isothermally to state 4 and finally compressed adiabatically until it reaches its initial state. Since the several reversible processes may be passed through in reverse order, the cycle itself is a reversible cycle.

324. Q.—What is the Rankine cycle?

A.—A steam cycle which is known as the Rankine, or as often termed, the Clausius cycle, is the most efficient

cycle which may be performed by steam without evaporating and condensing the working fluid within the engine cylinder, it has been adopted as the standard of efficiency with which the efficiency of all other cycles may be compared.

The engine is assumed to have no clearance, and the walls of the cylinders to be non-conductors of heat. Steam is admitted from a boiler from point A, Fig. A324, to point B, the expansion being isothermal and the boiler and steam pipe being a part of the expansion



A324. THE RANKIN CYCLE

chamber. At B cutoff occurs, and the steam contained in the cylinder expands adiabatically to the pressure of the exhaust pipe, as shown by line B-C, some of it condensing during the process. At the end of this expansion, the steam is discharged into the exhaust pipe at a constant back pressure, line C-D representing this process of isothermal discharge. Line D-A represents the rise in temperature and pressure without change in volume which results when the inlet valve opens.

325. Q.—Give examples of indicator diagrams from throttling and automatic cutoff engines at varying loads.

A.—In a plain slide valve engine with a throttling governor, the events of the stroke take place at the same point in the cycle at all times and the speed is regulated by the admission of more or less steam through the governor valve to meet the requirements of the load. Therefore, an indicator card from such an engine would look like Fig. A325-1.

With an ordinary single-valve high-speed engine, the valve travel and position of the eccentric on the shaft with reference to the location of the crank pin, are under control of the shaft governor which changes all events of the stroke to meet the changes of load and pressure, but these changes are more noticeable in cutoff and compression than admission and release, therefore a card traced through several revolutions of such an engine would look like Fig. A325-2.

FIG. 1

FIG. 2

A325-1. DIAGRAM FROM THROTTLING ENGINE, LOAD VARYING.
2. AUTOMATIC CUTOFF DIAGRAM, LOAD VARYING

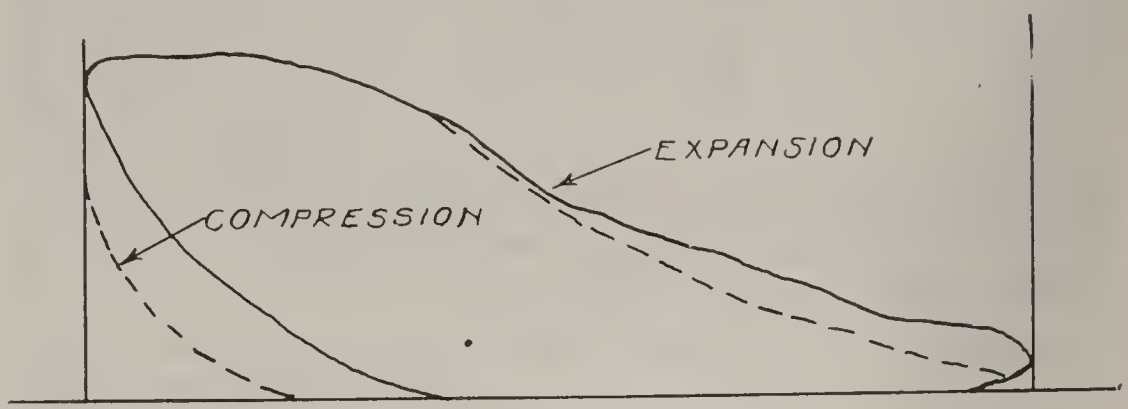
326. Q.—WHAT ENGINE defects will an indicator diagram show?

A.—Correctly attached and manipulated, the steam engine indicator will show improper valve setting and steam distribution, and excessive steam leaks past piston and valves, but slight leaks so often balance that they do not show on the diagram.

327. Q.—DRAW AN indicator diagram showing too much compression and leaky admission valve.

A.—Compression is a question of speed, in addition to the other things, and what would be enough compression in one instance might be too much or not enough in another. The effect of a leaky admission valve is to raise the expansion line and the terminal pressure as

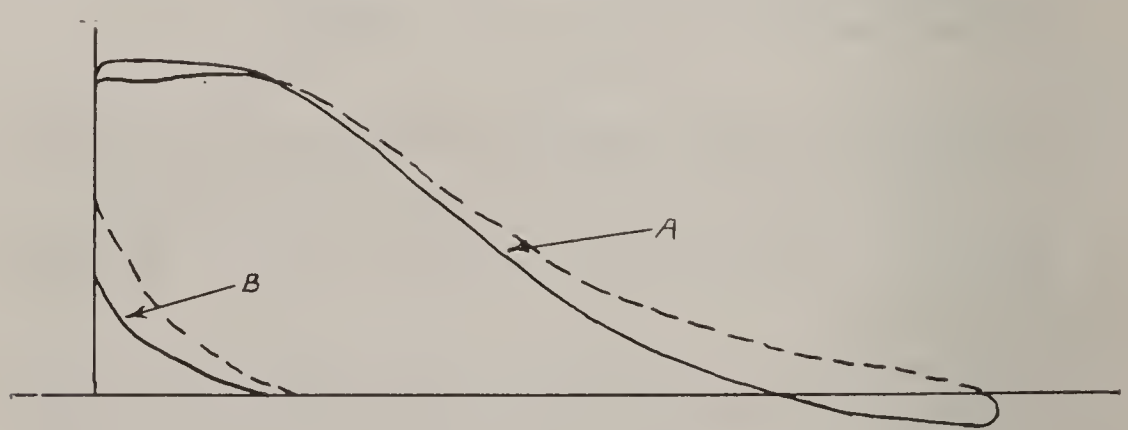
shown in Fig. A327. The dotted lines show normal compression and expansion.



A327. DIAGRAM WITH EXCESSIVE COMPRESSION AND STEAM VALVE LEAKING

328. Q.—How would a leaky piston show on an indicator card?

A.—If the piston leaked, any steam blowing past it into the end that was exhausting would not affect the card from that end as the excess steam would pass out at the exhaust valve. But as soon as the exhaust valve closes and the steam is being compressed, part of this passes the piston which lowers the compression line where it merges into the admission line, B, Fig. A328.



A328. EFFECT OF PISTON LEAKAGE

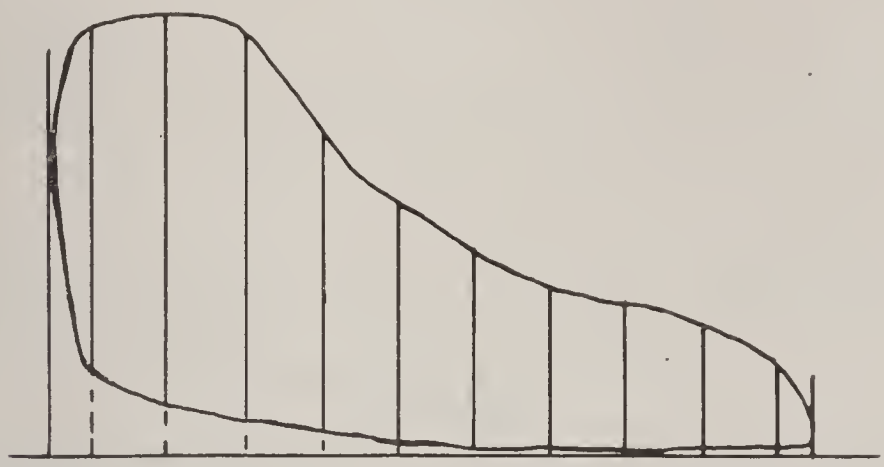
On the other hand, as soon as the steam valve closes, part of the steam leaking past the piston causes the expansion line to fall off to such an extent that it loops below the atmospheric line as shown, A, Fig. A328. This same effect could be caused by a leaky exhaust valve but if this were the case, the defect would likely show up more on one end than on the other, as it is hardly possible that both exhaust valves would leak the same amount.

329. Q.—If the admission line on an indicator card inclined inward from the perpendicular, what would it indicate in the valve setting?

A.—"Leaning in" of the admission line of an indicator card is usually caused by late admission. If the steam valve is late in opening, the piston gets into motion before steam is admitted, and the admission line curves inward. In some instances, however, the valve has had plenty of lead and opened before the return stroke was completed, but the exhaust valve was so late that not only did it not close for compression, but it did not close until the piston had started well on the forward stroke, so that steam was blowing direct into the exhaust and the pressure was not maintained. As the exhaust closed, the pressure was increased, but the piston was moving away so rapidly that the admission line never became vertical.

330. Q.—How would you figure horsepower from an indicator card?

A.—To find the horsepower of an engine is best and easiest done by means of a planimeter, by which



A330. DIAGRAM SPACED FOR MEASURING MEAN EFFECTIVE PRESSURE

you can get the mean effective pressure directly from the card, and then using the formula P L A N ÷ 33,000, which will give you the horsepower.

In this formula, P = the mean effective pressure in pounds per square inch; L = the length of stroke in feet; A = the area of the piston in square inches; N = the number of working strokes per minute.

If, however, no planimeter is at hand, the mean effective pressure can be found by computation. This

is done by dividing the diagram in the manner shown in Fig. A330.

The object here is to obtain the mean height of the 10 rectangular sections into which the diagram may be assumed to be divided. Each of the ordinates may be measured by the scale corresponding to the spring used, and the pressures so obtained added together; the result, divided by 10, will give approximately the mean effective pressure per square inch throughout the stroke, after which you can use the above formula to get the horsepower.

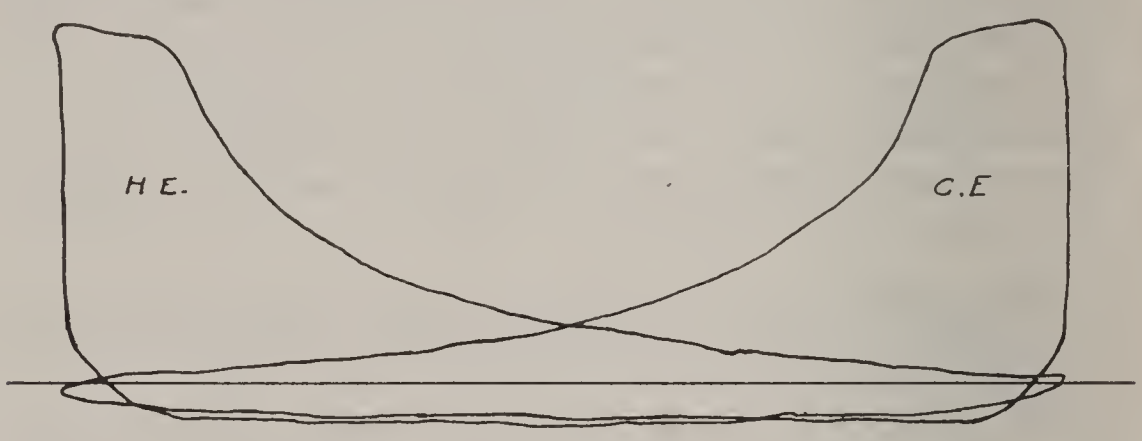


FIG 1

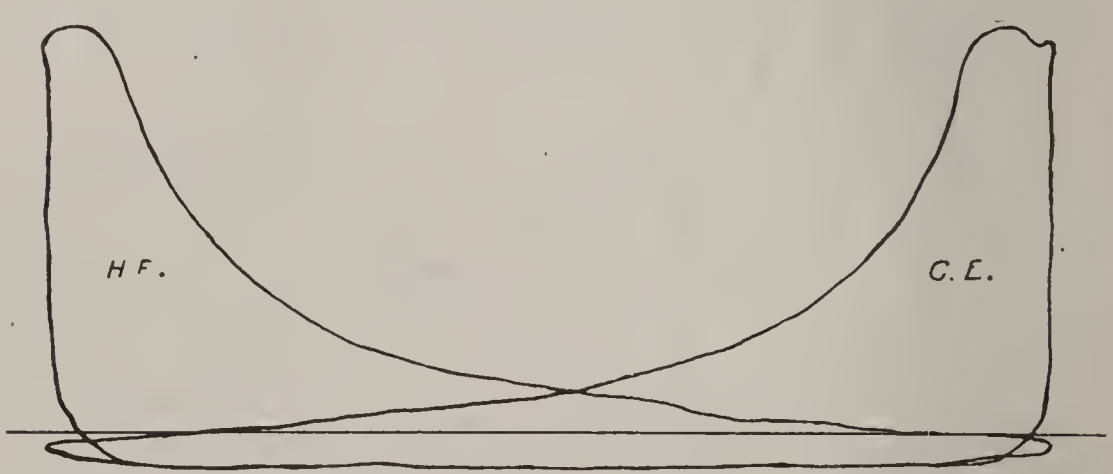


FIG. 2

Q. 331-1 AND 2 CORLISS CARDS

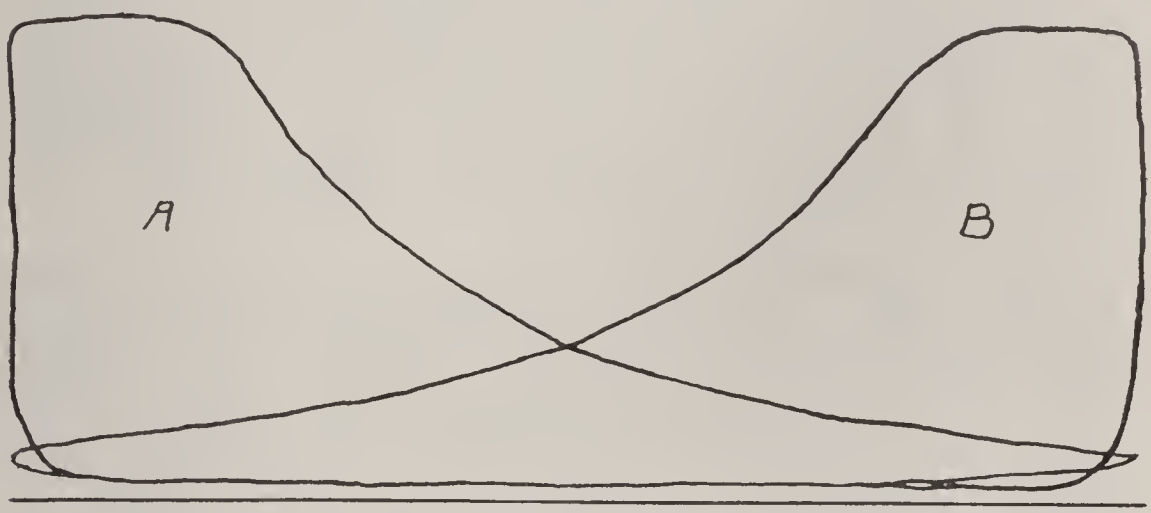
331. Q.—DIAGRAMS. Q. 331-1 and Q. 331-2 are from a 28 by 60-in. Reynolds-Corliss engine running condensing under a varying load. How can they be improved?

A.—The cards are well balanced as far as the distribution of the load is concerned. If these cards are fair representatives of usual operating conditions and no other steam using machine prevents, the pressure can be dropped about ten pounds or so.

The cutoff can take place a little later and secure better results. You might reduce the exhaust lap a little and advance the eccentric on the shaft a small amount. This will make compression take place at the same place as at present, and make release earlier. It will also increase the lead which will tend to carry the steam line a little higher up to the point of cutoff.

It seems that a higher vacuum ought to be maintained, but as this is in all probability outside of the engine itself no adjustment of the engine would better it.

332. Q.—A CORLISS engine, 20⅛ by 42 in., has been rebored from 20 by 42 in. Valve chambers have been rebored and fitted with new valves. Since this the



Q. 332. DIAGRAM FROM POUNDING ENGINE

engine has pounded on the head end, and keying up crank and crosshead pin boxes does not remove the trouble. Diagrams are given herewith. Can the difficulty be caused by late opening of the exhaust valves?

A.—You refer to the engine pounding on the head end, since it was rebored, but you overlooked designating on the card which was the head end. We have, therefore, identified them as "A" and "B" for the sake of reference.

Taking the card as a whole, we are inclined to think that the principal cause of the trouble is on account of the lateness of the events controlled by the eccentric. Move the eccentric ahead (in the direction in which the engine runs), say, about ⅝ in. as a trial. This will make all events earlier (except cutoff), which will give you release and compression at about the right place.

It will also give a little earlier admission, and these changes ought to go a long ways towards making the

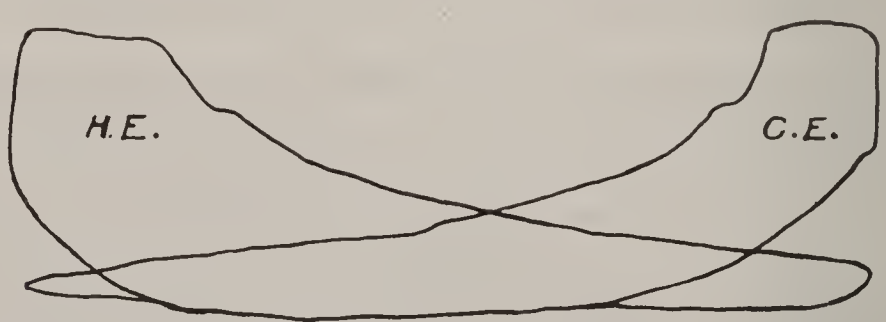
engine run quiet. At present the admission on the end "B" is later than on the end "A" which may be due to faulty adjustment of the steam rods, or else an "over travel" on the wristplate. If the latter, adjust the connection between the wristplate and the eccentric where means are provided, so that the travel will be equal.

If this is now all right, adjust the steam rod for "B" so that the admission will be the same on both ends.

Before going too far with adjustments on an engine, be sure that you are doing the adjusting in the right place, for engines have the unpleasant trick sometimes of making the sound audible at a place where it is not.

Lateral motion in valves will sometimes lead one to look for the trouble in the crosshead or the main bearing. Or a troublesome crosshead can sometimes be traced to the cylinder foundation plate and be caused by vibration of the plate when the dashpot seats.

333. Q.—Diagram Q. 333 was taken from a Russell 4-valve engine, 15 by 20-in., running at 170 r.p.m. on 75 lb. gage pressure, with 60-lb. spring. What causes the crook in the expansion lines? The engine is old, but on examination 6 mo. ago, the cylinder was in good shape.



Q. 333 DIAGRAM FROM RUSSELL ENGINE

A.—While the card submitted is not as good perhaps as one would wish it, yet the defects are not great and with a little care and thought you can restore the engine to normal conditions.

However, there are some things that may be encountered that we cannot foresee and which will give you a little trouble in applying the proper remedy. For instance, you say the engine is an old one, and if it has been run in the condition the card indicates for any length of time, in all probability the valves

have become "seated" and have worn a slight shoulder at the extremes of travel.

Possibly the pins in the valve gear are worn flat. If so, the pin and keyhead connections should be looked over and made right. This repair work must precede the readjusting of the valve travel.

The compression at the head end seems to be about right for your engine and the conditions under which it is operating, but the late release and early compression on the crank end show that the error is one that can be corrected by the reach rod. Adjust this so that you will have about the same compression as at the head end and you will then have a release that will permit a rapid exhaust of the steam.

When these adjustments are made, the little hook in the compression line on the crank end will disappear. Equalizing the load between the two ends of the cylinder is just a question of adjusting the main valve stem of the cutoff valve (the one under control of the governor). With a little care, taking cards in the meantime, the load can be equally distributed. This inequality was possibly caused by continued wear and adjustment of the eccentric rod brasses without making due allowance for it on the rod or valve stem.

As lead seems to be lacking, at least enough for the conditions, advance the main eccentric on the shaft, say, about ¼ in. This ought to bring the admission line about straight and not leaning inwards as shown on the card. The present location is all right for a much higher steam pressure than you are carrying.

That little jog in the expansion line is more likely a fault in indicating than in valve adjustment. A little friction in the piston of the indicator, which might be caused by water, or lack of lubrication would cause it, or it might be caused by the cord climbing over a knot on the pulley of the reducing motion. See that the cord leads onto the drum and pulleys smoothly.

334. Q.—An Atlas, 4-valve engine, running at 100 r.p.m. on 90 lb. gage pressure gave the diagram Q. 334. How can the head end card be made right?

A.—The card submitted from your Atlas 4-valve engine seems to be a sort of "abnormal monstrosity," inasmuch as the line at the head end usually occupied by the lines of compression and admission, has a reverse

action which is not in keeping with the sympathetic motion that is universal between the indicator and the motion of the piston.

The line referred to shows an indenture which cannot be accounted for through any action of the steam or valves but it is probable that the card for the head end was taken with too little tension in the drum spring, and consequently with a loose string which whipped at the end of the stroke enough to cause the reverse action of the drum.

Q. 334 DIAGRAM FROM ATLAS ENGINE

As for the head end card itself, the compression is barely evident at the point below the arrow. At this point admission takes place, much too early, causing the loop at the top of the diagram.

Since the crank end card looks so good, we are of the opinion that there is little if any disarrangement of the connections between the eccentrics and the wristplates, but this cannot be stated definitely. Possibly when you begin to eliminate the head end troubles you will find that some slight changes will have to be made in the eccentric rods or hook rods.

There is a slight steam leak at the crank end, but the head end shows considerable wire drawing. Better test out your cylinder for leaks in both piston and valves and if steam is any object, have repairs made. It does not take long for a ton of coal to squeeze through a small leak in a valve.

For present improvements give the head end more compression and less lead until you get about the conditions you have now on the crank end. Take cards as you go along and do not make a single move until you are satisfied what results you want to obtain and how to go about it to get them.

As you say that the engine runs but 100 r.p.m., we assume that it is not a small one, and therefore the

compression at the crank end might be reduced a little also.

335. Q.—THE INDICATOR cards shown in Figs. Q. 335-1 and 2 were taken from a 19 by 37 by 20 in. angle compound engine driving a centrifugal pump in a suction dredge. The high-pressure cylinder has a piston valve

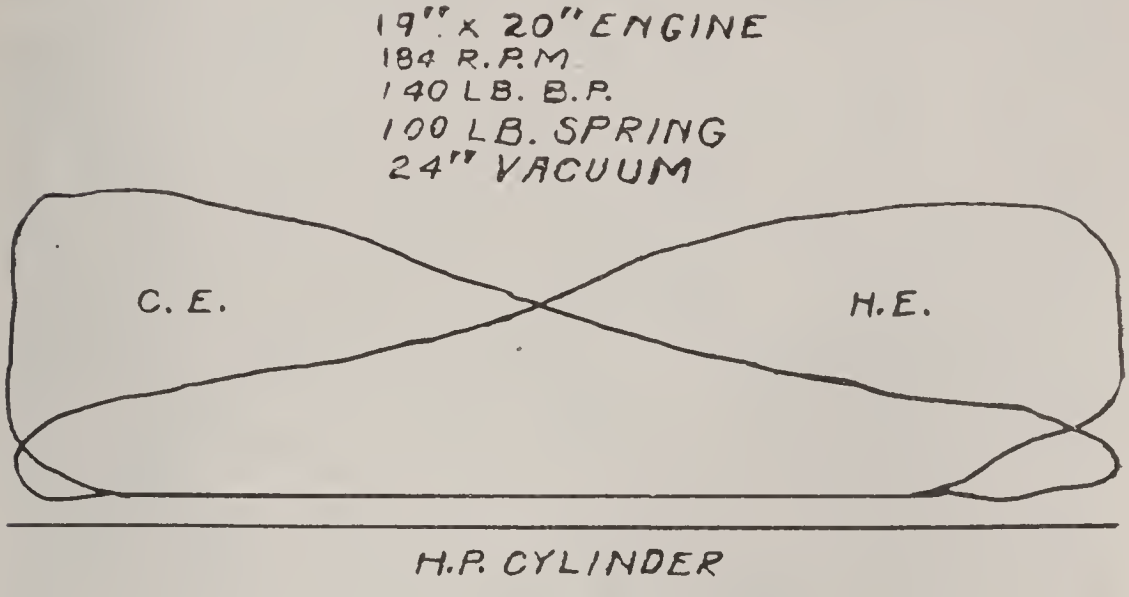


FIG. 1

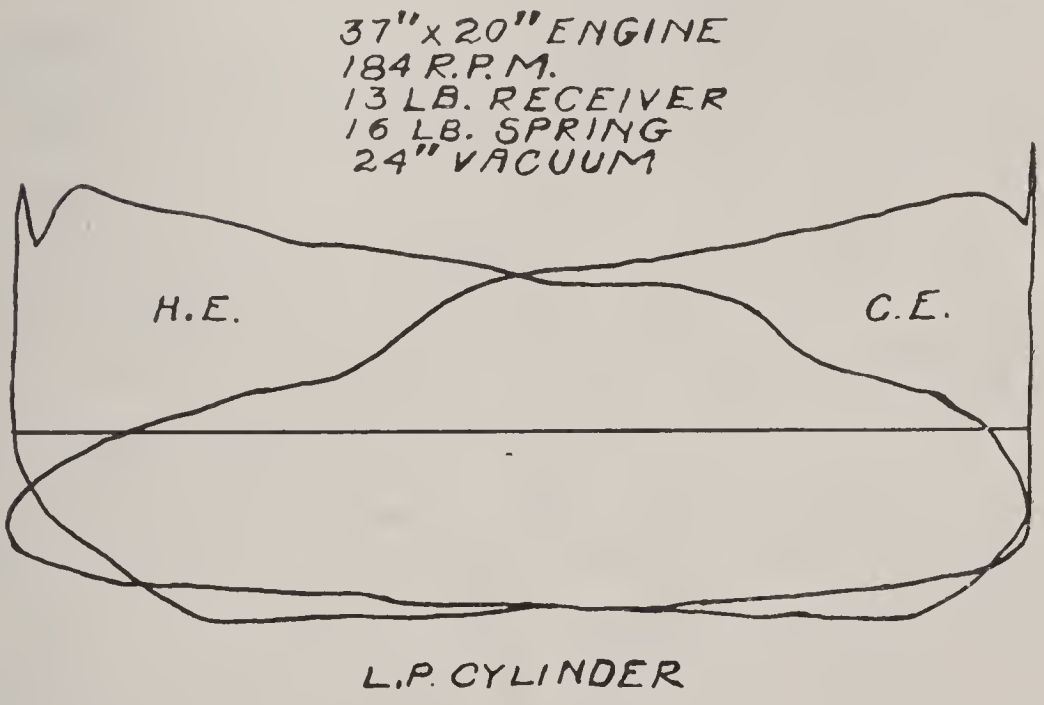


FIG. 2

Q. 335-1 AND 2 DIAGRAMS FROM ANGLE COMPOUND ENGINE

and Meyer cutoff working inside of the valve; the low-pressure has a slide valve. The piston rings in the high-pressure are carried on a bull ring with restricting lugs on the bull ring to let rings out only so far. The low-pressure has one sectional ring 1 in. thick by 4 in. wide, held in place by springs behind it, against the piston, the whole clamped down by a follower plate. The high-pressure cylinder is 1/64 in. out of round and the low-pressure 1/32 in. We have a great deal of trouble keeping crank pins from wiping the metal when turning 180 r.p.m. Both connecting rods work on one pin 21 in. between webs of crank with a 1-in. ring in center to keep brasses apart making the brasses each 10 in. long, of which ¾ in. is taken up with fillets. The pins are 10 in. diameter and oil from outside through the brasses. The so-called brasses are steel, babbited with Challenge metal. The bearing of brasses on pins is about 67 sq. in. for each. The oil used is heavy Red Star engine oil, Standard Oil Co. We did use XXXX marine engine oil, but burned a set of brasses once a month. Two journals carry all the weight and have a marine thrust of the horseshoe type.

A.—Your problem presents some features that are interesting. It is hard to prescribe anything definite without knowing more of the conditions, but we will try to offer a few suggestions that might assist you in overcoming your difficulties.

Of course, any engine driving a suction dredge is subjected to considerable wear from flying grit and dust, which is unavoidably present. This will help to make a bad matter worse.

Although you do not say so, we presume that the speed is controlled by the throttle, for while the speed indicated on the cards is 184 r.p.m., the initial pressure in the high-pressure cylinder is from 100 to 110 lb., while the boiler pressure is noted to be 140 lb.

If this surmise is correct, it accounts for this large drop in pressure. The high-pressure cards show a very indistinct point of cutoff, although it appears to take place at about 30 per cent of the stroke.

It appears to us that your trouble is due more to late admission than to anything else. In fact the several events of the cycle in the high-pressure cylinder seem to be late, and the compression is greater at the head end than at the crank end.

We would suggest that you advance the eccentric of the main high-pressure valve, say, about ⅜-in. on the shaft as a trial. This ought to make the admission line straight instead of inclined as at present, and increase the compression. Then adjust the main high-pressure valve stem so as to decrease the compression at the head end and increase it at the crank end.

If the Meyer cutoff can be adjusted by hand, reduce the point of cutoff, and let the steam expand more in the high-pressure cylinder. By a little study, you may be able to bring the initial pressure up to within a few pounds of boiler pressure. The high-pressure valve leaks also, which helps the trouble along. True it is caught by the low-pressure cylinder, but it is getting past the high without doing any work. In fact, the low-pressure valve leaks also, so that possibly what leaks into the low, about balances what leaks out, so there is a loss both ways.

We take it that the steam to and from the low-pressure cylinder is handled by a single slide valve. If this is the case, it would not hurt any to advance this eccentric a small amount also, say, about the same as the main high.

You see, at present, owing to the late admission in the high, the piston almost drags at the beginning of the stroke, and when admission does take place, there is considerable thrust to it, which is pretty hard on the babbit in the crank pin boxes. By advancing the eccentrics as directed above, this evil will be eliminated for the piston will have full initial pressure on it at the moment of beginning the stroke.

Now to go down to the cranks and look around a little. You say that the crank pin for each cylinder is 10 by 10 in., but that the bearing of the brasses is but 67 sq. in. If this is so, then the brasses are relieved too much at the joint and for this reason:

The area of the 19-in. high-pressure cylinder is 283 sq. in. Suppose, just to make the calculation easy and simple, that the initial pressure is 100 lb. Then there is a pressure on the piston at the beginning of each stroke, of 28,300 lb., and to take up this strain on the crank pins, there is but an area of 67 sq. in. which makes the pressure on the pin about 421 lb. per sq. in. It takes a good bearing to stand this punishment.

While it is good practice to relieve crank pin boxes

at the joint to prevent "gripping" should they warm slightly, yet they should not be relieved so far back as to reduce the effective area to anywhere near what it is in your case. If it is relieved about 1 in. from the joint, and allowing ¾ in. at each end of the pin for fillets, this will give you an effective area of about 90 sq. in., which reduces the pressure on the pin to about 315 lb., which is considerable.

To obtain the best results with a heavy bearing, it should be babbited around a mandril about ⅜ in. smaller and then peened with a 2 or 3-lb. ball peen hammer. After this, bore it accurately, making it about 0.003 in. larger than the pin. Scrape it to an absolute fit with the pin. Relieve it about an inch back from the joint, and bevel the edges so that the oil will feed in easily.

In cutting oil grooves, the writer has found that the letter "X" method is the best, cutting a groove also at the cross of the "X," and running lengthwise of the pin to within half an inch of the end of the brass. This insures a pocket of oil at all times.

Your cylinders seem to be worn out of round considerable. There is no remedy for this but reboring and that usually means new steam rings and possibly bull ring and follower also.

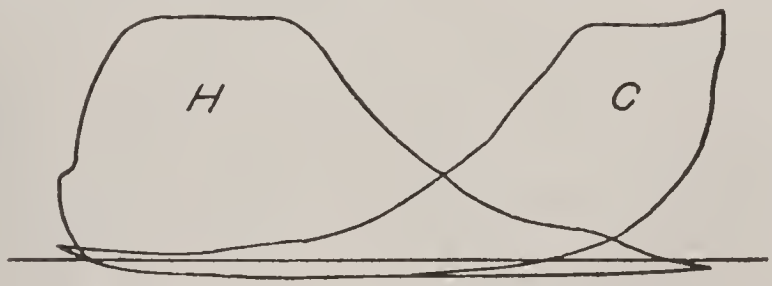
336. Q.—Stroke of an engine is 40 in. and arm of the reducing motion 7 ft. long. How far from the point of suspension of the arm should the indicator cord be attached to give a diagram 3 in. long?

A.—The total length of the reducing motion arm will be to the distance from the point of suspension to the attachment of the indicator cord as the length of stroke is to the length of the cord. All values must be in feet or in inches. Working out the proportion of 84:x::40:3 gives us 6.3 in. as the distance from point of susepension to the indicator cord.

337. Q.—What effect would it have on the valves of a Corliss engine if the rocker did not vibrate equal distance each side of a plumb line?

A.—While it is desirable to have the rocker arm (or carrier arm, as it is more frequently called), oscillate equally on both sides of the vertical center line, yet if the arm be long, as is common on some makes of en-

gines, a difference of ½ in. is not noticeable on the valve adjustment, provided the throw of the wristplate is equal. In fact, sometimes one cannot do otherwise than to throw the carrier arm out a little in order to secure proper throw at the wristplate. With a tandem Corliss engine, it frequently happens that the theoretical perfect adjustment of the rods and connections is dealt a sad blow in order to strike a happy medium where the best all-round conditions prevail.



A337. RESULT OF WRISTPLATE OVERTRAVEL

With the short, stubby type of carrier arm that is used on some of the recent designs of Corliss engines, a slight deviation will affect the steam distribution more or less provided no other adjustments are changed.

Let us assume a single eccentric Corliss engine with a right and left eccentric rod, that this rod for some reason was unscrewed, say, half a turn, and that the threads were four to the inch. Then the eccentric rod would be ¼ in. too long and the carrier arm and the wristplate would overtravel towards the head end just that much.

The effect on the steam distribution would be as follows, if it was correct before the accident: Head end late admission, crank end early admission, and greater load on head end. On the exhaust,—head end early release and late compression; crank end, late release and early compression. The card shown in Fig. A337 (exaggerated, of course, to bring out the features), would be about what the indicator would show.

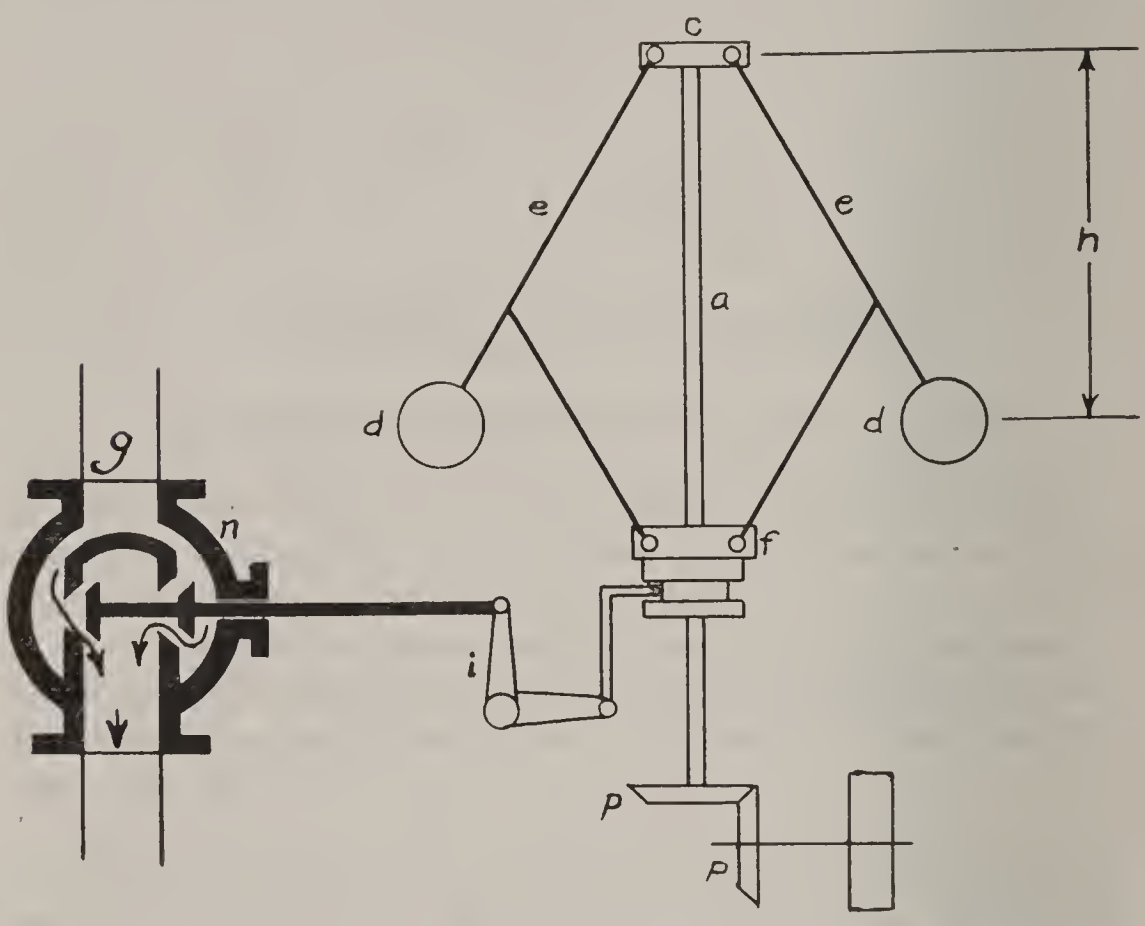
Engine Governors

338. Q.—What is the principle on which the flyball governor operates?

A.—Let us assume a vertical rod *a*, Fig. A338, connected to the main shaft of the engine by means of a belt operating through a pulley and two beveled gears *p* and *p*, so that the rod *a* revolves at the same speed as

the engine shaft. Suspended from pivots in a block *c* at the upper part of rod *a* are two rods, *e, e,* on the extremities of which are fastened the weights *d, d.*

In accordance with the principles of centrifugal force and gravity, as the rod *a* increases or decreases in speed, the weights *d, d,* will rise or fall. Pivoted at *e, e* are two rods also pivoted to the collar *f,* which is



A338. DIAGRAM OF FLYBALL GOVERNOR

free to rise and fall upon the rod *a.* Any change in speed of the engine shaft will cause a corresponding change in the speed of the rod *a,* also a change in the position of the weights *d, d* and in the vertical position of the collar *f.* If now the main steam pipe *g,* supplying the engine be considered as having a valve at *n,* the valve rod being connected to the bell crank *i,* which in turn is connected to the collar *f,* it will be seen that as the speed of the engine increases, due to a decrease of load, the collar *f* will rise and, through the various levers and the valve *n,* will partially close the opening of steam pipe *g.* On the other hand a decrease in speed, due to an increase of load, will cause the collar *f* to drop, thereby opening the valve *n.* In the one case steam at lower pressure will be admitted to the cylinder and in

the other at higher pressure, thus providing for the change of load, and maintaining a constant engine speed.

339. Q.—How does a throttling governor control the supply of steam to an engine? How the shaft governor?

A.—A throttling governor does not control the amount of steam going to an engine, but controls the pressure that is delivered to the steam chest, as described in question 338.

The throttle governor regulates the speed by changing the initial pressure. The steam is wiredrawn through a partially closed valve. The shaft governor regulates the speed by changing the cutoff without changing the initial pressure, thus regulating the amount of steam entering the cylinder each stroke.

340. Q.—With the ordinary Corliss engine governor, or flyball governor, what effect, if any, would it have on the speed of the engine if you doubled the weight of the balls themselves—not the "dead weight" or the counter-weight?

A.—Putting governor balls twice the weight in place of those already in use, will not change the speed of a Corliss engine, when a simple pendulum governor is used. The centrifugal force will be doubled but also the gravity force will be doubled. The governor will be more powerful, that is, it will overcome more easily any friction in the rods which run to the cutoff cams, but the extra weight in the balls will bring extra strain on the governor arms and would probably be a dangerous proposition.

The formula for this type of governor is,

$$h = \frac{35{,}192}{n}$$

where n = number of revolutions a minute

h = height of the pendulum in inches (h, Fig. A338).

For the loaded pendulum governor, the formula is:

$$h = \frac{W + L}{W} \times \frac{35{,}192}{n}$$

W = weight of each ball in pounds

L = weight of center load in pounds

h = height of pendulum in inches.

From this it will be seen that doubling the weight of the balls of a loaded governor will affect the speed, the amount depending upon the center load. If you double the value of W, the speed remaining the same, the value of h is decreased; this means a shorter cutoff with consequent decrease in speed. When a balance is finally reached the result will be a decrease in speed of the engine.

341. Q.—How does a pendulum governor control the speed of a Corliss engine?

A.—The pendulum governor is connected by means of rods to the knockoff cams, which disconnect the steam valves from the wristplate. When the governor changes its position, these rods change the point at which the knockoff cams act.

342. Q.—How does the automatic work on a Gardner governor and a Waters governor, and how do you test them to see that they are working?

A.—Gardner governors, which are of the flyball type, are made in three classes. Class "A" governor is of the gravity action type, and has an automatic safety stop and speeder. The centrifugal force of the pendulum balls is opposed by the resistance of a weight on the stop lever. The action of the stop is accomplished by permitting a slight oscillation of the shaft bearing which is supported between centers and held in position by the pull of the belt; a projection of the lower part of the shaft bearing supports the fulcrum of the speed lever. If the belt breaks or slips off the pulley, the support of the fulcrum is forced back, allowing the fulcrum to drop, instantly closing the valve.

Class "B" governor is a combination of the spring and gravity action. It is furnished with speeder lever but has no automatic stop. In its action, the centrifugal force of the pendulum balls operates against the resistance of a coiled steel spring, enclosed within a case and pivoted on the speed lever by means of a screw; the amount of compression on the spring can be changed so as to give a wide range of speed. The third type of Gardner governor is the spring governor which is similar to the other two, but more sensitive.

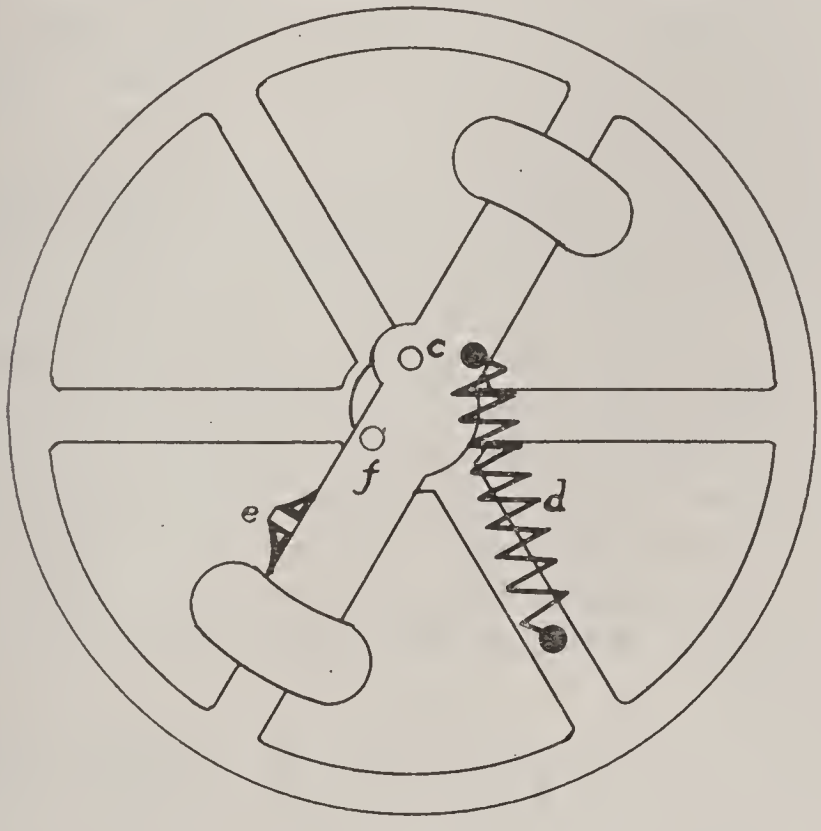
In the Waters governor, the balls are mounted on flat-blade springs which are bent into shape and at-

tached to the spindle in a horizontal position. The balls are small and light and revolve at high speed.

To test a governor of the flyball type, turn the governor pulley by hand which, if it turns freely, shows that the working parts do not bind. Then raise the balls or compress the springs to see that the valve and stem work easily through the packing; then examine and replace the belt. Start up the engine and test the speed with light and heavy loads. It should hold constant within 4 per cent.

343. Q.—What is an inertia governor and how does it work?

A.—The Inertia Governor, Fig. A343, consists essentially of a bar with a weight at either end pivoted at



A343. DIAGRAM OF INERTIA GOVERNOR

the pin *c*. A spring *d* tends to hold the weight bar in the starting position against the stop *e*. The eccentric rod is connected to the pin *f* which takes the place of the eccentric. As the speed increases, the centrifugal force of the weighted ends increases until it overcomes the tension of the spring *d*, and the bar swings about its supporting pivot *c*, changing the position of the pin *f* relative to the center of the shaft, and producing the same effect as produced by the shifting of the eccentric. If there should occur a sudden change of load the

weights will tend for a moment to continue at the same speed at which they were traveling. They may, therefore, run ahead or lag behind. The result of this action is to change the position of the eccentric pin in its relation to the center of the shaft. In such a governor the inertia of the cross bar and weights materially supplements the centrifugal force and increases its sensitiveness.

344. Q.—Why is a shaft governor more economical than a throttling governor?

A.—The shaft governor is more economical than throttling, because it varies the cutoff as the load changes and thus cuts down the amount of steam taken by the engine, while giving full expansion of the steam.

345. Q.—How can you tell that an engine with a shaft governor will not run away without taking the valve chest cover off?

A.—To determine whether or not this is the case, without opening the valve chest, place the engine on dead center and open the throttle valve, having, at the same time, blocked the governor arms in their extreme or full speed position. If any steam passes through, the valve has lead, which it should not have. The governor must, when in its full speed position, practically blank the ports. Should the engine have any lead at all when the governor is in its full speed position, it will be impossible for the governor to cut off the steam supply entirely, in which case, the engine is bound to run away.

346. Q.—If the tension spring of a shaft governor should break, what would the engine do?

A.—The purpose of the spring in a shaft governor is to maintain such a balance against the centrifugal force of the governor weights that for different speeds of the engine the governor arms will take definite positions. With no springs, that is, when the spring breaks, there is nothing to resist the centrifugal force and the weights will swing out to the stops; the effect of this action upon the valve motion depends upon the type of gear or eccentric, but in any case, the steam would be cut off as much as it is possible to do with the governor, which would cause a very decided reduction in speed or the engine would stop entirely.

347. Q.—If the heavy weight of an inertia governor broke off outside the spring, what would happen?

A.—With a Rites inertia governor the tendency is for the heavy end of the governor bar to lag behind which it does until the restraining action of the spring just balances this tendency. Changes of load and pressure alter these positions under ordinary operating conditions, but if the heavy end of the bar were to break off outside of the spring connection, there would be no restraining influence on the spring and it would, therefore, draw the eccentric into full throw position so that the engine would speed up, and if the load were light enough, the speed would be limited only by the endurance of the machine.

348. Q.—For an engine driving a generator, which is better, a sluggish or a sensitive governor?

A.—In the case of a sluggish governor we find, assuming the engine to be running with a given load at some definite speed, that an increase of load will cause a noticeable drop in speed, due to the delayed action of the governor. Then if no change in load occurs after this the governor will gradually reach its proper position where the engine will get just the right amount of steam to maintain the proper speed. In a like manner, should the load drop off the engine would tend to speed up, gradually, however, returning to the correct speed as the governor responds.

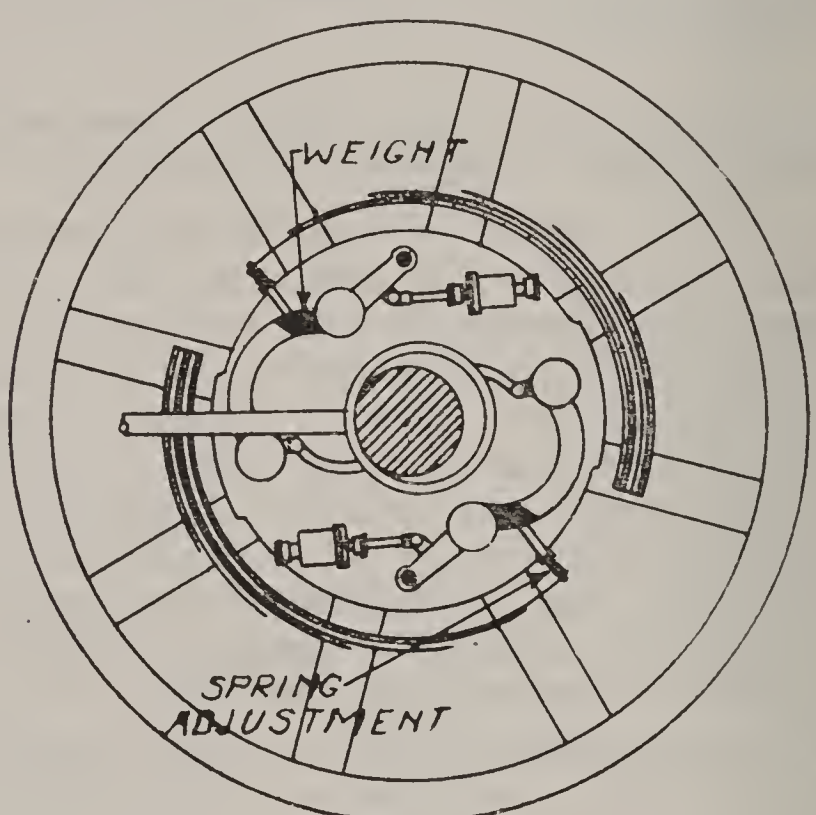
Let us now assume having an engine with an oversensitive governor. An increase of load will cause the governor to respond so quickly and to such a degree that the engine will run for the instant at a comparatively high speed, when the governor, due to its sensitiveness, will shut off the supply of steam, causing the engine speed to drop noticeably. This rapid rise and fall of speed, or "hunting" as it is termed, has a bad effect on the generator voltage, showing up badly on the electric service, particularly if used for lighting. Therefore, if it became a matter of choosing between the two evils, we should prefer the sluggish governor.

No fast rule or rules can be given as to how a governor should act, as this depends largely upon service requirements. It is more a matter of cut and try until you obtain that degree of sensitiveness which will result

in a minimum of speed variation with a maximum load variation.

349. Q.—If at varying loads, a McIntosh & Seymour engine runs slow or fast, how do you adjust the governor to make it run steadily?

A.—With a McIntosh & Seymour engine governor, the tension pins between the springs and the centrifugal weights are arranged to telescope so that they can be adjusted to secure proper sensitiveness. By lengthening these pins, the governor can be made to regulate more closely, and by shortening them, over-sensitiveness can



A349. MC INTOSH & SEYMOUR GOVERNOR

be removed. The speed at which the engine will run can be raised or lowered by reducing or increasing respectively the small lead weights provided for that purpose in holes in the centrifugal weights, although this adjustment does not alter the sensitiveness of the governor. Any adjustment of the sensitiveness of the governor, however, will change the speed. Thus, after the pins connecting the springs and the centrifugal weights are shortened to diminish the sensitiveness at varying loads, any decided change in speed must be remedied by an adjustment of the weights.

350. Q.—An engine with pendulum governor runs at 100 r.p.m. and has the pulley on the shaft 8 in. diameter. To change speed to 90 r.p.m., what diameter pulley should be used?

A.—In order to slow down the engine, it is necessary to use a larger engine pulley to drive the governor, so that the governor may be run at the same speed with the engine running slower. The increase in diameter will be in direct proportion to the decrease in engine speed. We would, therefore, have $8 \times 100 \div 90$, which gives 8.9 in. as the diameter for the new engine pulley.

351. Q.—How do you change the speed of a Corliss engine?

A.—To change the speed of the governor the size of the pulley on the governor shaft should be changed. To find the required size of the new pulley multiply the diameter of the pulley on the governor shaft by the desired speed of the engine and divide the product by the present speed of the engine. The governor will run at about a certain height for a given governor speed, and in order to change the engine speed, the ratio of the governor speed to that of the engine must be changed. The speed of the engine may be changed by changing the counter weight on a weighted governor, but except for a slight change this is not advisable as it tends to throw the governor out of proper balance.

352. Q.—What might cause an engine to "hunt" immediately after the load was thrown off?

A.—Several things may make the engine "hunt" or race for several seconds when the load goes off. Excessive friction of the cutoff device caused by accidental straining of some of the rods or connections will tend to retard the governing action of the governor. These parts must be absolutely free from strain of any kind.

On most governors as used on Corliss engines, there is a gagpot or a vessel in which a plunger works in oil to steady the speed. There is some device by means of which the oil in the pot is allowed to pass from one side of the piston to the other through a small regulating valve which may be either outside of the pot or in the plunger itself.

By trial, this opening can be set at a point where it will allow the oil to pass through just fast enough

to secure good regulation. If this setting is varied either way, racing or hunting will be the result.

As a general thing, a light machine oil is the best to use in the gagpot and sometimes it is necessary to add kerosene also in order to secure the best results.

The earlier types of governors for Corliss engines were of the flyball type without counterpoise. These governors are not so sensitive to speed changes as those with a dead weight counterpoise and lighter weight balls which depend on a high rotative speed to secure close regulation.

When the load is suddenly thrown off, this governor does not respond rapidly enough and the engine speeds up before the governor can control it. The excessive speed naturally increases the speed of the governor several revolutions when the extra centrifugal force throws the flyballs too far outward where they revolve in a new plane, in which position cutoff takes place too early, and the engine slows down.

This oscillation above and below normal speed goes on for several seconds until the engine again comes under control. It is entirely possible that the racing is caused by a combination of these causes.

353. Q.—What might be the cause of a variation in the speed of a high-speed engine?

A.—If the speed of a high-speed engine varied, it would, no doubt, be due to governor trouble. This might result from insufficient lubrication, improper adjustment, or wear. The trouble might also result from excessive friction of the valve, or between the eccentric and strap.

354. Q.—If an engine was running at 100 r.p.m., and you wanted 10 more r.p.m., what would you do to make this change?

A.—This depends on the type of engine, but on any engine the governing mechanism would have to be readjusted. On the Corliss engine this could be done by decreasing the ratio of the speed of the governor to that of the shaft by using a smaller pulley on the governor shaft; or adding weight to the counterweight would increase the speed. On an engine using a shaft governor the change could be made by moving the weight on the weight arm or by changing the tension

of the spring. Changing the size or weight of the weights would be the best way for a change of 10 r.p.m., as changing the tension of the spring so much might make the governor regulation poor and it would not operate close to the desired speed.

355. Q.—If a slide valve engine ran too fast how would you decrease the speed?

A.—If a slide valve engine is running too fast, the governor may be adjusted either as to the spring tension or the setting of the governor shutoff valve. If this is impossible, the governor pulley may be changed so that for a given speed of the engine, the governor will run at a higher speed.

Power and Steam Consumption

356. Q.—What is the rule for figuring the horsepower of a steam engine?

A.—The power developed by a steam engine is determined by means of the following formula:

Hp. = P L A N ÷ 33,000, in which P is the mean effective pressure in the cylinder (m.e.p.); L, the length of the stroke in feet; A, the area of the piston in square inches, and N, the number of working strokes per minute (twice the number of revolutions in a double-acting engine). For accurate results the mean effective pressure is determined by measurement of the indicator diagram.

357. Q.—The horsepower constant of an engine is 3.833; the mean effective pressure is 40 lb. per square inch. What is the horsepower of the engine?

A.—The horsepower constant of an engine is the power developed for one pound of mean effective pressure in the cylinder. This constant multiplied by the mean effective pressure gives the horsepower developed; in the example given $3.833 \times 40 = 153.32$ hp.

358. Q.—An engine making 75 r.p.m. with a m.e.p. of 40 lb. develops 100 hp. If the m.e.p. is raised to 45 lb. and the speed to 80 r.p.m. what horsepower would be developed?

A.—The power developed by an engine is in direct proportion to the mean effective pressure and the speed. Therefore, dividing 100 by 40 and multiplying it by 45,

dividing the product by 75 and multiplying by 80 gives us the increased horsepower, 120.

359. Q.—Which is greater brake horsepower or indicated horsepower?

A.—Brake horsepower is less than indicated horsepower, because the former is the net power at the engine shaft, while the latter is the power developed in the engine cylinder. A portion of the indicated horsepower is used in overcoming friction, and does not, therefore, appear as useful energy.

360. Q.—In simple and in compound engines, how is the ratio of expansion found? Also the number of expansions?

A.—The ratio of expansion is found by dividing the final volume of the steam, just before exhaust opens, by the volume at cutoff, both volumes including the clearance. For a simple engine both volumes are for the one cylinder. For a compound engine, final volume is taken for the low-pressure and cutoff volume for the high-pressure.

If cutoff in the high-pressure is at ¼ stroke, calculate the piston displacement and divide by 4; to this add clearance volume and the sum is volume at cutoff. Similarly calculate the low-pressure piston displacement at exhaust opening and add clearance volume, to get final volume. Divide this last by the cutoff volume, and the quotient will be the ratio of expansion.

This is the same as the number of expansions.

361. Q.—In a compound engine, high-pressure cylinder 14 in. diameter and low-pressure cylinder 30 in. diameter, with 36-in. stroke, clearance 5 per cent in each cylinder, cutoff occurs at ⅓ stroke; what is the total ratio of expansion by volume?

A.—The volume of the high-pressure cylinder at cut-off $= 0.7854 \times 14^2 \times 12 + 0.05\ (0.7854 \times 14^2 \times 36) = 2124$ cu. in.

The volume of the low-pressure cylinder at release is $0.7854 \times 30^2 \times 36 + 0.05 \times 0.7854 \times 30^2 \times 36 =$ 29,869 cu. in. $29{,}869 \div 2124 = 14 +$ expansions.

362. Q.—What is the number of expansions in a Corliss engine 20 by 40 by 42-in., cutting off at ¼ stroke in each cylinder?

A.—The number of expansions which take place in this compound engine is 16.

This is calculated as follows: The volume of high-pressure cylinder to cutoff equals 3300 cu. in. The volume of low-pressure cylinder at release equals 52,750 cu. in.; 52,750 ÷ 3300 = 16, number of expansions.

363. Q.—How WOULD you make a Corliss engine develop more power, speed and steam pressure to remain the same?

A.—The only way to increase the power would be to put in a new cylinder and piston of larger diameter, which would probably need new piston rod, connecting rod, cross head and crank.

364. Q.—How CAN a 50 hp. engine be increased to 75 hp., boiler pressure remaining the same? What parts would you have to look out for?

A.—To increase the horsepower of an engine from 50 to 75 without increasing the boiler pressure, it would be necessary to increase the speed 50 per cent. However, in a case of this kind, you would have to know that the engine would stand this extra speed, as the fly wheel might not be strong enough with a good factor of safety, and the pins and bearings also might not stand the additional speed. Cutoff might be lengthened somewhat, depending on the type of engine.

365. Q.—How DOES the horsepower of an engine increase by advancing the cutoff from one-fourth to one-half stroke? A table I have gives the following: One-fourth cutoff mean pressure per pound initial, 0.59; terminal, 0.25; one-half cutoff, 0.84; terminal, 0.50.

A.—In order to answer your question in a comprehensive manner, it will be necessary for us, first, to take up the formula employed to determine the indicated horsepower of any reciprocating type of steam engine. This is P L A N ÷ 33,000; where P is the mean effective pressure, in pounds per square inch; L, the length of the stroke, in feet; A, the area of the piston, in square inches, and N, the number of strokes per minute.

With a given speed, and as the values of L and A cannot, in any given engine, be varied, we find that this formula is directly proportional to the value of P, the mean effective pressure of the steam within the cylin-

der. The value of this factor is, in turn, proportional to the point at which cutoff occurs.

Let us assume that we have a boiler pressure of 100 lb. Under these conditions, and the engine cutting off at one-quarter stroke, the value of P will be equal to 0.59×100 or 59 lb. (from table showing ratio of mean to initial pressure). If, however, the cutoff is advanced to one-half stroke, the value of P will be increased to 84 lb. per square inch, and as a consequence, the horsepower developed will be increased proportionately.

366. Q.—What is the most accurate way of determining the steam consumption per indicated horsepower of a steam engine?

A.—With a constant load for some given interval such as, let us say, 15 min., determine the amount of steam used by means of a reliable steam-flow meter; multiply this by 4 for the consumption per hour at this 15-min. rate. During this time take a number of indicator cards from which work up the indicated horsepower, using the average obtained. Divide the steam consumption in pounds per hour by the number of indicated horsepower and the quotient will be the consumption per indicated horsepower-hour.

367. Q.—How may the steam accounted for by the indicator be obtained from the indicator diagram? What is the value of such a determination?

A.—The water rate or steam consumption of an engine may be obtained from the indicator diagram, as follows:

Let p = mean effective pressure; l = length of stroke in feet. a = area of piston in square inches; $a \div 144$ = area in square feet; c = clearance percentage of stroke; b = per cent of stroke at point where water rate is to be computed; n = number of strokes per minute; 60n = number of strokes per hour; w = weight of 1 cu. ft. of steam having a pressure as shown by the diagram corresponding to that at the point where water rate is required; w′ = that corresponding to pressure at end of compression.

Number of cubic feet per stroke $= l\,(b + c)\,a \div 14{,}400$.

Corresponding weight of steam per stroke in pounds $= l\,(b + c)\,aw \div 14{,}400$.

Volume of clearance $= l\ c\ a \div 14{,}400$.

Weight of steam in clearance $= l\ c\ a\ w' \div 14{,}400$.

Total weight of steam per stroke $= [l\ (b + c)\ a\ w - l\ c\ a\ w'] \div 14{,}400 = l\ a\ [(b + c)\ w - c\ w'] \div 14{,}400$.

Total weight of steam from diagram $= 60\ n\ l\ a\ [(b + c)\ w - c\ w'] \div 14{,}400$.

Indicated hp. $= p\ l\ a\ n \div 33{,}000$. Hence the steam consumption per hour per indicated horsepower is:

$$S_c = \frac{60\ n\ l\ a\ [(b + c)\ w - c\ w'] \times 33{,}000}{14{,}400\ p\ l\ a\ n}$$

$$= 137.5\ [(b + c)\ w - c\ w'] \div p$$

The value of a determination such as this is largely dependent upon instrumental and personal accuracy.

368. Q.—How does the accuracy of the steam consumption figured from the indicator card, compare with figures secured by condensing and weighing the exhaust steam?

A.—Condensing and weighing the exhaust steam or use of a steam flow meter is the only accurate way to get steam consumption. Figuring the steam consumption from the indicator card is only an approximation, and the accuracy of the work depends entirely on the success in estimating what to allow for condensation of steam during admission. This will vary from 10 to 30 per cent, depending on the condition of the steam, piston speed, boiler pressure and the size of the engine.

Roughly, an allowance of 20 per cent may be made above that shown by the indicator diagram volume, and for ordinary conditions of power plant engines, this will be sufficient.

369. Q.—To determine the amount of steam used per kilowatt-hour would you divide the total amount of steam consumed by the product of the number of amperes times the number of volts, divided by 1000?

A.—If you are using direct current, your method of finding steam per kilowatt-hour—that is, dividing the steam consumed per hour by (amperes times volts divided by 1000)—would be correct. If you are using alternating current, it would be necessary to multiply the amperes by the voltage and by the power factor of the system, to get the total watts.

370. Q.—In a horizontal engine, which stroke takes more steam, the forward or backward?

A.—Due to the fact that the area of the crank end side of the piston is reduced, or is smaller than that of the head end side because of the piston rod, the forward stroke requires more steam than the backward stroke, cutoff being the same for both strokes.

371. Q.—A prony brake has a length of 5 ft. from the center of the shaft to the point of attachment to a platform scale. When power is applied, the scale balances at 100 lb. and when not applied at 37 lb. The engine makes 150 r.p.m. What is the brake horsepower?

A.—The increase in scale reading due to the load is 100 minus 37 or 63 lb. In one revolution of the wheel this travels through a distance, or is overcome through a distance, equal to the circumference of a circle whose radius is 5 ft. This circumference would be equal to 3.1416 times 10. Multiply this circumference by 63, the weight, by the speed, 150, and divide by 33,000, and it gives the horsepower, 90.

Flywheels

372. Q.—What is the safe speed for the rim of a flywheel to travel?

A.—The maximum velocity of a flywheel rim is ordinarily taken as 6000 ft. per minute.

In calculating the safe speed for a cast-iron flywheel, assuming a tensile strength of 10,000 lb., and using a factor of safety of 10, this formula is used:

$$V = 1.6 \sqrt{\frac{\text{Tensile strength} \div \text{factor of safety}}{\text{Weight per cubic inch}}}$$

$$= 1.6 \sqrt{\frac{1000}{0.26}}$$

giving 100 ft. per second as the safe speed. For cast-steel, assuming a tensile strength of 60,000 lb., and a factor of safety of 10, the safe speed is $1.6 \sqrt{\frac{6000}{0.28}} =$ 233 ft. per second.

373. Q.—Please give an idea how to obtain the theoretical weight of a flywheel in a short way.

A.—Thurston gives for engines with automatic valve gear,

$$W = 250{,}000 \frac{A\,S\,p}{R^2\,D^2}$$

in which A = area of piston in square inches
S = stroke in feet
p = mean steam pressure in pounds per square inch
R = revolutions per minute
D = outside diameter of the flywheel in feet.

Another formula given by Thurston for ordinary forms of noncondensing steam engines, having a ratio of expansion of from 3 to 5, is

$$W = \frac{a\,A\,S}{R^2\,D^2}$$

In this formula, a ranges from 10,000,000 to 15,000,000, averaging 12,000,000, while the other letters denote the same as above.

374. Q.—What weight of flywheel is required for a 16 by 36-in. Corliss engine running 90 r.p.m. The wheel is to be 10 ft. diameter and 19 in. face.

A.—Using the second formula of answer 373 and substituting the values given:

$$W = \frac{12{,}000{,}000 \times 201 \times 3}{(90)^2 \times (10)^2} = 8940 \text{ lb.}$$

For the hub and spokes, about 35 to 40 per cent more is added.

Supposing the rim to be laid out flat, its outer surface then contains:

$$10 \times 3.1416 \times \frac{19}{12} = 49.7 \text{ sq. ft.}$$

Allowing cast iron to weigh 465 lb. per cubic foot, the rim should contain:

$$\frac{8940}{465} = 19.2 \text{ cu. ft.}$$

The thickness of the rim is then:

$$\frac{19.2}{49.7} \text{ or } 0.387 \text{ ft.} = 4\tfrac{5}{8} \text{ in., approximately.}$$

375. Q.—What is the rule for strain on the bolts in the joint of a flywheel rim? Diameter of wheel, size of rim and speed are given.

A.—Let W = weight of the rim in pounds
R = radius of the rim in feet
n = revolutions per minute
$g = 32.16$
v = velocity of the rim in feet per second
$= 2 \times 3.1416\ R\ n \div 60$

The centrifugal force of the whole rim radially is:

$$F = W\ v^2 \div g\ R = W\ (2 \times 3.1416\ R\ n)^2 \div (g\ R \times 60^2)$$
$$= 0.000341\ W\ R\ n^2$$

The resultant of half of the radial force, acting at right angles to a diameter, tends to disrupt one-half of the wheel from the other half, and is resisted by the sections of the rim at the ends of the diameter, or by the bolts at the joints. It is this force that the bolts in the joints of a two-piece rim are required to resist.

The resultant of half the radial force taken at right angles to a diameter is:

$$F \div 6.2832$$

Hence the tensile strain on the cross-section of the rim is:

$$S = 0.000341\ W\ R\ n^2 \div 6.283.2$$
$$= 0.00005427\ W\ R\ n^2$$

The weight W of a rim of cast iron 1 in. square in section is $2 \times 3.1416\ R \times 3.125 = 19.635\ R$ pounds; whence strain per square inch of sectional area of rim, $S_1 = 0.0010656\ R^2n^2 = 0.0002669\ D^2n^2 = 0.000027\ V^2$, in which D = diameter of wheel in feet, and V is velocity of rim in feet per minute. $S_1 = 0.0972\ v^2$, if v is taken in feet per second.

For wrought iron:

$$S_1 = 0.0011366\ R^2n^2 = 0.0002842\ D^2n^2 = 0.0000288\ V^2.$$

For steel:

$$S_1 = 0.0011593\ R^2n^2 = 0.0002901\ D^2n^2 = 0.0000294\ V^2.$$

For wood:

$$S_1 = 0.0000888\ R^2n^2 = 0.0000222\ D^2n^2 = 0.00000225\ V^2.$$

Compound Engines

376. Q.—What is the advantage of running an engine with a condenser?

A.—Using a condenser with an engine gives a low back pressure, which increases the power of the engine more than the amount of power required to maintain a vacuum. The possible heat efficiency of an engine depends on the ratio of the difference in temperatures between the steam at admission and exhaust, divided by the admission temperature—all these temperatures being in degrees absolute. As the back pressure is lowered by the use of a condenser, the difference in temperatures is increased without increasing the divisor in this efficiency expression.

377. Q.—Is a compound engine more economical than a single cylinder engine? What is the saving?

A.—The most economical type of steam engine depends entirely on the use for which it is to be installed. A compound engine will generally have less steam consumption per horsepower-hour, but in case exhaust steam is wanted for industrial purposes, a single cylinder engine may give the most economical plant. For a steady load with high-pressure steam, the compound engine is most economical, but with variable load and low-pressure steam, it is not.

The saving in steam consumption to be made by compounding is variable, and is due principally to the reduction in the initial condensation and re-evaporation in the cylinder, by keeping the cylinder walls at a more even temperature. A much greater saving can be made at a high pressure than at a low pressure. The speed, terminal pressure, point of cutoff and speed of the engine also have some effect on it. With high pressure, say 150 lb., a saving of 10 to 15 per cent would be probable without adding a condenser.

378. Q.—If by running condensing, a saving of from 20 to 30 per cent may be realized, will compounding add to these figures? If so, how much?

A.—Compounding will increase the saving in the case of an engine run condensing. The amount of saving depends on the initial boiler pressure carried and the vacuum carried, but will run probably from 8 to 10 per cent.

379. Q.—Is it a good plan to compound a "Simple" engine; under what conditions would it be a paying investment?

A.—The radiation losses from a compound engine are greater than those of a simple engine, and very often the mechanical losses are increased by compounding.

The tendency then in a compound engine is to increase the radiation loss and increase the mechanical losses. On the other hand, compounding decreases the thermodynamic losses by decreasing the range of temperature in each cylinder. With low pressure and a small number of expansions, the single cylinder engine is more economical than a compound engine, but with high pressure and a larger number of expansions, the compound engine is more economical than the simple.

For pressures under 100 lb., the single cylinder condensing engine is generally more economical than the compound, but for pressures above 100 lb., the compound engine is usually the more economical.

In the case of non-condensing engines, the compound engine does not show any economical advantage until the pressure reaches 150 lb. In general it may be said, that the higher the pressure and the larger the number of expansions, the greater the economy of the compound engine.

380. Q.—If the diameter of the high-pressure cylinder of a triple-expansion engine is given, the initial pressure, the cutoff in the high-pressure cylinder and the terminal pressure, how would you find the size of the other cylinders?

A.—A common rule for proportions of volumes of cylinders of a triple expansion engine (Kent) is as follows:

The ratios of the high to the intermediate and of the intermediate to the low are equal to the cube root of the number of expansions, the ratio of the high to the low being the product of the two ratios, that is, the square of the cube root of the number of expansions.

For example, if the number of expansions is 15, the cube root of 15 is 2.47 and 2.47 squared is 6.08, thus the cylinder volume ratios would be approximately, 1 : 2.47 : 6.08. In practice this varies somewhat. The ratio of the diameters of the cylinders would be approximately

the square root of the ratio of the areas, if strokes are equal.

The total number of expansions is found by dividing the absolute initial steam pressure by the absolute terminal pressure. Thus, if the absolute initial pressure is 175 lb., and the absolute terminal pressure is 6 lb., the total number of expansions is $175 \div 6 = 29.1$.

381. Q.—Diameter of the high-pressure cylinder is 16 in., boiler pressure 175 lb., cutoff at ¼ stroke. What are the sizes of the other cylinders and of the receiver?

A.—Using the method of A380, and taking the number of expansions which you would probably have as 16, you would have a ratio, then, of 1 to 2.52 to 6.38, and working this out, we get, as the diameter of your high pressure cylinder, 16—the intermediate 25.4, and the low pressure as 40.3. This work has been done on the slide rule, and in round numbers the cylinders may probably be made 25 and 40 in. in diameter. The volume of the receiver for triple expansion engines, where the cranks are placed 120 deg. apart, and the intermediate piston leads the high pressure, is very small; probably one-half that of the cylinder would be sufficient.

382. Q.—How is cutoff changed in the low-pressure cylinder?

A.—The manner of changing the cutoff on the low-pressure cylinder will depend on the type of engine. On the Corliss engine with both cylinders controlled by the governor, the length of the governor rods on the low pressure side could be adjusted. In some compound engines the governor acts on the high pressure side only, and the cutoff on the low pressure side is fixed. In this case the length of the knockoff rods would be changed by hand adjustment.

383. Q.—How many expansions of steam are there in a compound engine?

A.—The total expansions in a steam engine of compound type is found by multiplying the number of expansions in the high-pressure cylinder by the ratio of cylinder volumes. Neglecting clearance—if you have cutoff at ¼ stroke in the high-pressure cylinder, there will be four expansions in that cylinder; and if the

ratio of cylinder volumes is 3, the total expansions would be 4 times 3 or 12.

384. Q.—What would be the terminal pressure in the low-pressure cylinder of a cross-compound engine if the cutoff in the high-pressure cylinder was at quarter stroke; allowing nothing for drop, and the cylinder ratio being four?

A.—In a cross-compound engine, with the high pressure cylinder cutting off at quarter stroke, and a cylinder ratio of 4, there will be $4 \times 4 = 16$ expansions between the initial and terminal pressures. Therefore, the terminal pressure would be 1/16 of the initial pressure.

385. Q.—Which is easier to regulate—compression of a cross-compound or a tandem-compound engine? Why?

A.—We presume that reference is made to Corliss engines in preference to other types. In a Corliss tandem-compound engine, no matter whether the engine is equipped with double eccentrics or not, there are more connections between the eccentric and the valve gear of the rear cylinder and in taking up wear the tendency is to distort the valve setting of this cylinder much more than would be the case on the cylinder next to the frame or on either cylinder of a cross compound.

If the side rod be fixed to the wristplate, it is usually provided with right and left connections to permit accurate adjustments, but if there is a releasing hook so that the rear cylinder gear can be disengaged and operated by hand, close adjustment is not possible as the hook rod must be turned one complete revolution or not at all, and it seldom happens that the wristplate travel is equalized with this kind of a device.

In such case it may not be possible to secure just as good adjustments on the exhaust valves as one would wish, and one must compromise between the two cylinders so as to secure as good all-around operating conditions as circumstances will permit.

It sometimes happens that one cylinder of a tandem might require more advance of the eccentric than its mate, and here one must compromise again, getting some of the desired results on the valve rods if necessary; but, at best, the cards will not be ideal.

On a cross-compound engine one has to consider but one cylinder at a time, and wristplate travel and advance

of the eccentric can be worked out satisfactorily without having to keep in mind the fact that there is another cylinder that will be thrown out of harmony.

As far as valve setting is concerned, a cross compound can for the time being be considered as two separate simple engines, and there are no complications beyond keeping in mind what each part is for and how best to make it perform its duty.

386. Q.—If the high-pressure cylinder of a compound Corliss engine broke down, how could you tell what pressure was needed to carry the same load on the low-pressure cylinder? Steam pressure 110 lb., vacuum 28 in., cylinder volume ratio 4 to 1.

A.—Assume that the cutoff is at ¼ stroke in the high-pressure cylinder, so that the total ratio of expansion is 16.

To find the mean effective pressure in the high-pressure cylinder, we divide the initial pressure absolute, which is 110 + 15, or 125 lb., by the ratio of expansion 4, and multiply this by (1 + the hyperbolic logarithm of the ratio of expansion). For the ratio 4, this logarithm is 1.39. We then have, $125 \div 4 \times 2.39 = 74.7$ lb., the mean forward pressure. From this we subtract the receiver pressure, which is $125 \div 4$ or 31.2 lb. absolute, leaving as the mean effective pressure in the high pressure cylinder 43.5 lb.

In like manner, the mean effective pressure in the low-pressure cylinder is $31.2 \div 4 \times 2.39$, which equals 18.65 lb., and from this we subtract the back pressure, which is the vacuum pressure, 0.98 lb., leaving a mean effective pressure of 17.67 lb.

Now, to get the same effect in horsepower, we shall have to have a mean effective pressure on the low-pressure cylinder equal to that when the engine is running compound, plus ¼ that on the high-pressure cylinder. Dividing 43.5 by 4, we get 10.9, which, added to 17.67 gives us 28.57 as the mean effective pressure needed when using the low-pressure cylinder alone. The mean forward pressure will be this amount plus 0.98, the back pressure, or 29.55 lb.

For our equation above, assuming cutoff still remains at ¼ stroke, in the low-pressure cylinder, we should have as initial pressure, $29.55 \times 4 \div 2.39$, which gives us 49.5 lb. as the initial pressure absolute, and from this

we subtract 15, giving us 34.5 lb. as the gage pressure at which the engine should be run.

387. Q.—Why is it necessary to have a receiver on a cross-compound engine? Is one necessary on a tandem-compound?

A.—In the cross-compound engine cranks are usually set at 90 deg. and the cylinders do not work simultaneously; steam exhausted from the high-pressure is held in the receiver before admitted to the low-pressure cylinder. For this reason a receiver between the high-pressure and low-pressure cylinders is necessary in order that the exhaust pressure may not build up on the high-pressure piston before admission to the low-pressure.

It is not necessary to have a receiver with a tandem-compound engine, for as the steam is exhausted from the high-pressure cylinder it immediately goes to the low-pressure cylinder.

388. Q.—Is there any way to determine the receiver pressure without using a pressure gage?

A.—The following method may be used to find the receiver pressure, but, if it is at all possible, a gage on the receiver will undoubtedly get more accurate results, as the formula given below is for isothermal expansion, and does not take into account the initial condensation; to be exact, you must know the clearance of the piston. It is also necessary to indicate the engine in order to find the terminal pressure.

The formula for isothermal expansion is:

$$P\,v = \text{a constant};$$

where P is the absolute pressure, and v is the volume. From which we get, that multiplying the terminal pressure absolute by the number of expansions in the low-pressure cylinder gives the initial steam pressure on the low-pressure cylinder or the receiver pressure absolute.

Receiver pressure, absolute, may also be found by multiplying initial pressure absolute by the high-pressure cutoff volume and dividing by low-pressure cutoff volume.

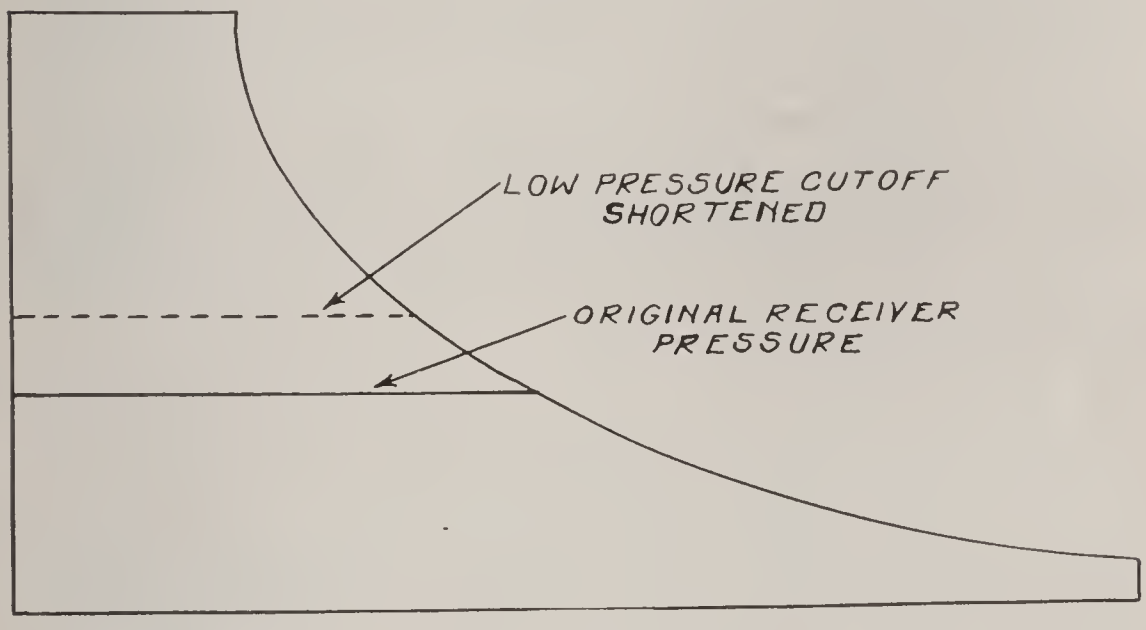
389. Q.—What will be the effect if the receiver of a compound engine has too high or too low a pressure?

A.—If the receiver pressure is too high in a compound engine, it will result in increasing the work on the low-pressure side, and decreasing that in the high

pressure. It will also bring an excessive strain on the parts on the low-pressure side. On the other hand, a pressure that is too low in the receiver will increase the work done by the high-pressure side, lessen the work done by the low-pressure side, and bring an increased strain on the parts on the high-pressure side.

390. Q.—How can more of the load be thrown onto the low-pressure cylinder?

A.—To throw the load on the low-pressure cylinder it is necessary to increase the mean effective pressure on that cylinder, and decrease it in the high-pressure cylinder. To do this, shorten the cutoff on the low-pressure



A390. EFFECT OF SHORTENING LOW-PRESSURE CUTOFF

cylinder. At first thought this does not seem right, but it is accomplished in this way. When the cutoff is shortened the low-pressure cylinder will take a smaller volume of steam from the receiver at each stroke. This increases the receiver pressure, consequently the back pressure on the high-pressure cylinder. With a given initial pressure the high-pressure cylinder at the increased back pressure will have a smaller mean effective pressure than with a low back pressure. But as the initial pressure in the low-pressure cylinder has been increased it will have a greater mean effective pressure than it did before the cutoff was changed. Increasing the vacuum in the condenser will also increase the load on the low-pressure side.

391. Q.—What is the absolute pressure in the condenser when the barometer reads 30.1 in. and the vacuum gage shows 28.5 in.?

A.—The absolute pressure will be the difference between the barometric reading and the vacuum gage. In your case, 30.1 minus 28.5 will give 1.6 in. of mercury. An inch of mercury is equal to 0.492 lb. per square inch, and 1.6 times this gives us 0.786 lb. per square inch as the absolute pressure.

Operation of Engines

392. Q.—How do you change the cutoff on a Corliss engine?

A.—The cutoff of a Corliss engine is automatically under the control of the governor, and can only be altered by changing the position of the governor, which may be accomplished by changing the steam pressure, changing the load, or changing the speed of the engine. The cutoff on the two ends may be changed in relation to each other, that is one end made late or early by changing the relative length of the knockoff cam rods to the governor.

393. Q.—How can you tell where a Corliss engine is cutting off while running?

A.—If the engine is running at full speed it would be impossible to get more than a rough estimate of the point at which cutoff takes place. To determine the cutoff make a mark on the governor post opposite a point on the vertical stem of the governor when running with the usual load. Shut the engine down and block the governor up to the mark, hook up the valves, and turn the engine over slowly by hand until cutoff takes place. Measure the distance the crosshead has traveled from the starting point. Divide the number of inches the crosshead moved to point of cutoff by the length of stroke in inches to get the percent of cutoff. Measure the other end in the same manner, as they may not be alike.

394. Q.—With heavy load on our Corliss engine the dashpots work all right, but as soon as the load goes off they will rebound when they drop and the grab claw has to drive them down every time. The knock-off cams are all right. Only one dash pot bothers at a

time, sometimes on one end of the cylinder, sometimes on the other.

A.—Your dashpot troubles may be caused by any one of several things. Naturally they work all right when the load is heavy, for then the lift is ample to provide the necessary vacuum under the plunger to insure a prompt return when the valve is tripped. If the lift is small, the plunger is not raised enough to create the proper vacuum to return the plunger to its place.

If lubricated with a heavy oil, this will retard the prompt return which is so much desired. Sometimes if the spring in the check valve is too strong it will prevent the escape on the return stroke of the air. which sucks in when the plunger is raised. Scant lubrication of the steam valves will also increase the trouble.

395. Q.—How would you know whether a Corliss engine was in proper condition to start, assuming that you were taking charge of a third-class plant that has been down or standing for some time?

A.—First, make a casual examination of the engine in a general way, observing its peculiarities, if there are any, and see that there are no loose connections.

Block the governor half way up in the slot and after setting at a dead center, turn the engine over until cutoff occurs and measure the distance the crosshead has traveled. Repeat this and measure the cutoff at the other end to determine whether the cutoffs are equal. See that the governor gag pot is properly adjusted, and that the pot is filled with light oil.

Next, drop the governor to its lowest position and see that the safety clips on the knockoff cams are adjusted so as to prevent the steam hooks from latching with the hook pins.

396. Q.—If you were running a Corliss engine non-condensing and you should add a condenser, what changes would you make in the valve adjustment?

A.—If the valves had been correctly adjusted for running non-condensing, no changes need be made in their setting for running condensing. But cutoff should come earlier unless the load is increased.

397. Q.—What would you do if the crankpin on

cross-compound engine on high-pressure side was running hot?

A.—As soon as it is found that the pin is heating, one should always render "first aid" in the form of increased lubrication, with the usual oil, or a little cylinder oil, or oil with a small quantity of pure graphite flour, powdered mica, or sulphur flour added, but not so much as to retard flow or to clog the oil grooves. Having done this, then look to the cause and a permanent remedy.

There are many reasons why any bearing may run hot, after continuous satisfactory performance. It may be from faulty lubrication or none at all, or some foreign matter entering the bearing, or in the case of a high-pressure crank pin, it may result from some slight disarrangement of the governing mechanism, thus reducing the load on the low-pressure cylinder and increasing it on the high-pressure cylinder and bringing the pressure on the high-pressure pin brasses up to a point where heating would result.

If it is found that this is the cause without a reasonable doubt, the only thing to do is to decrease the point of cutoff in the low-pressure cylinder with reference to that of the high-pressure cylinder, thus increasing the receiver pressure (which is the initial pressure for the low-pressure cylinder), and increase the back pressure on the high-pressure cylinder. A pound or two increase on the receiver gage makes considerable difference in pressures on pin bearings.

398. Q.—A Utica No. 9 35-hp. engine with slide valve and adjusting cutoff valve is governed by a No. 2½ Pickering ball governor. There is a pound in the crank pin just after it leaves center on the return stroke. The pin is loose in the brasses at this point but is tight for the rest of the revolution. Engine bed and crosshead are level, also the shaft, and the shaft is as nearly square with the cylinder as I can make it. The crank disc is 21 in. diameter and rolls 3/32 in. out of line in a revolution. Does this cause the pin to get out of line with the brasses? The engine vibrates on the cylinder end, causing the joints in the exhaust line to leak.

A.—We should judge from your inquiry and the description of your engine that it has seen considerable service and that at some time in its career it has not

had as good care as it should have had, for troubles such as you describe are not caused by anything but a continuation of these conditions during a period of several years.

In a case of this kind, it is best to "start all over again" and place the engine in as good shape as possible, for to repair one part and not another will not be satisfactory.

You should remove the piston and connecting rod and also the crosshead if necessary, and run a fine line through the cylinder and slide, set absolutely central, and with this as a starting point, you can readily learn the true condition of affairs.

Perhaps we misinterpret what you say in regard to the disk rolling 3/32 in. out of line, but it may be a little lateral motion in the shaft, or it may be that the disk was not pressed onto the shaft square, in which case the crankpin would be out of square with the shaft and also with the center line of the engine.

If it is lateral motion, some means must be found to reduce this to just a running fit, which can be done by drilling a few holes, say, ½ in., in the end of the bearings back of the babbit on the end next to the eccentric, and filling these with babbit, letting it project an eighth of an inch or so.

File or scrape this down until there is no lateral motion or just to a running fit. The design of your main bearing may be such that you will have to equalize this addition on both ends of the bearing shells, as sometimes the bearing is made with a tongue and groove, or a dowel pin which prevents the shells from moving laterally, and in case the wear is on both ends of the box babbiting on the eccentric side only will draw the disk away from the center line which would have a bad effect on the crank and crosshead pins. You will have to determine this yourself.

Having looked after the bearing and placed it in as good condition as possible, rotate the shafting until the pin is close to the line, say, about 1/32 in. Then, making sure that the disk is snug up against the bearing, caliper from the line to the end of the pin, preferably with a pin gage. Now rotate the shaft until the crank pin is close to the line at the other center, and caliper again.

If there is any difference, the outboard bearing must

be moved forward or back, as the case may be, until the distance is the same at each place. Then you can be sure that the shaft is square with the center line of the engine. See to it that the shaft is level.

While you have the line in the engine, also caliper from the edge of the disk to the line both at the line and at the opposite side and with the pin in both positions for measuring for squaring the shaft as described above. You can soon tell if the disk is out of square with the shaft.

Before testing the pin for squareness with the disk, one must know that the pin is round, which we believe is not the case, as you say that just after leaving the out center, the brasses are loose on the pin, but are tight at all other points of the revolution.

You know that the wear on a crankpin of a non-reversing engine is all in one place as the thrust at both centers takes place on the same part of the pin. Possible the shaft being out of square prevents this looseness from being noticed at the opposite center.

Even if the pin is out of round, however, there is little wear on the side next to the shaft, and you can use a steel square across the disk to test the squareness of the pin. If it is square with the pin but is not parallel with the center line your trouble is either a sprung shaft or a disk bored out of true, or else fitted with a key that has a top-and-bottom fit.

If the trouble is serious, the only remedy is to put in a new pin that is parallel with the shaft, and let the disk wabble.

As a further check for a sprung pin or shaft, with the crosshead adjusted to a snug fit, disconnect the connecting rod at the crosshead and after withdrawing it a little ways, draw up on the crank pin boxes tight, and note if the rod is thrown to one side or the other of its position in the crosshead.

Try this on both centers or as close as is possible to the center, making a careful note of results as you go along.

Possibly your trouble is aggravated by unequal valve setting, or by shoulders worn in the cylinder if the piston rings do not travel to the counterbore. This is frequently the case in small engines, and sometimes is found even in large engines.

399. Q.—What are the causes of knocking in an

engine bearing and what remedies should be used to stop it?

A.—The principal cause of pounding in engine bearings is lack of adjustment to compensate for wear. No matter how profuse lubrication may be, wear naturally occurs and adjustment must be made from time to time. On the other hand, one should not be continually tinkering with the engine bearings, as some engineers are prone to do, for bad results will surely follow.

If an engine is properly erected, the shaft will be square with the center line of the engine, and if the bearings are properly adjusted, the engine should operate quietly.

As the main bearing wears, adjustment is made from time to time, which naturally crowds the shaft towards the cylinder, but as the outboard bearing cannot be adjusted to compensate for the wear in the main bearing, the shaft will soon get out of square. With an engine of this design, the shaft should be squared up and shims added behind the rear quarter box so as to hold the shaft in the same position as it was when erected.

Should the engine be belted forward, the wear on the outboard bearing will be in the same direction, continually, while the rear quarter box of the main bearing will have its usual wear, owing to the intermittent thrust of the piston. This arrangement soon throws the engine out considerably with consequent pounding.

When the shaft is out of square, the crankpin and crosshead-pin boxes will slap from side to side, causing noise, and a continuance of this operation will cause the brasses to wear in the rod ends and even after the shaft has been squared up, these boxes will continue to pound unless shimmed up. Shimming strips may be sweated onto these boxes, or if the wear is so small that very thin shims must be used, plane off ⅛ or 1/16 in. less the wear and sweat on a piece of brass this thickness and scrape the brass to a perfect fit.

Of late years, engine builders have given more attention to engine adjustment and the main and outboard bearings are usually provided with adjustment both ways from the squared center line of the shaft. Sometimes either the main or outboard bearing, or both, is provided with a bottom wedge adjustment in order to keep the shaft level.

A cause for pounding little understood and, in fact,

ignored by many engineers, is loose babbit. How frequently we see bearing shells which have been melted out for rebabbiting, where the anchor holes are but an ordinary hole drilled in depth not more than the diameter of the hole.

Such a method cannot be criticized too severely. If for any reason anchor holes or slots are not cast in the shell, and it is necessary to drill them, the holes should be generous in diameter and depth, if the stock will stand it. Let us suppose an instance where the shaft is, say, 12 in. in diameter, and that the slot cores "fell out" when casting, leaving the shell wholly or partially devoid of any method of holding the babbit. Anchor holes should be at least ¾ in. in diameter and fully as deep, and at the bottom, holes about ⅜ in. should be drilled at an angle so as to form a "root" to the babbit.

These large holes should be spaced not more than 2 in. apart (less would be better), and should be slightly countersunk at the top, for if a sharp edge is left, the anchor may easily break off in operation.

Another cause, or, rather, presumed cause, of pounding in engine bearings, is a noise having its origin in another place, but which seems to be in the crankpin or main bearing. It occasionally happens that a crosshead shoe will become loose, especially if it be of the non-adjustable kind. By this is meant that type of shoe which is bolted directly to the crosshead itself, shims being inserted from time to time as required. To do this, it is necessary to remove the connecting rod and rotate the crosshead until the shoe comes out at the side of the bore, where it can be removed and shims inserted.

If the retaining screws become loose in operation, the shoe is not gripped tightly and the continued reversal of motion soon causes a little lost motion, which grows rapidly. If allowed to become too bad, machine off where wear has occurred and rivet on a steel shim, afterwards fitting the parts snugly and renewing the bolts attaching shoe to crosshead, using a size necessary to make a good job of it.

Lateral motion of the shaft is one cause of an annoying pound in the main bearings of engines. It is necessary to have at least some play here, but occasionally it is too much. Sometimes a wobbling wheel will cause lateral motion, or a wide band wheel which, though it may balance perfectly, could it be tested on gigantic

balancing rolls, might be heavier on one side of the rim than the other, this defect in the two halves of the wheel being on opposite sides of the rim.

If the engine is direct connected to a generator, it is possible to have the stator so perfectly set that the rotor moves in practically a neutral path and no magnetic influence is present to hold the shaft steady either against the crank or the collar. The remedy in a case like this is to move the stator laterally, preferably away from the main bearing, a small amount, in which case the magnetic influence will hold the crank constantly against the bearing.

If the shaft be moved too far, the thrust of the crank against the bearing will be so great as to cause heating.

400. Q.—How may one best balance a steam engine of the reciprocating type?

A.—The question of balancing an engine is rather an intricate one. Vibration comes from the stopping of the reciprocating parts, the piston, piston rod, cross-head and connecting rod—and their reversal at the end of the stroke.

There are two forces that have to be counteracted, the horizontal force of inertia of these parts, and the vertical force of inertia of the connecting rod swinging up and down. The horizontal force can only be counteracted by a heavy weight on the crank disk or a fan-tail weight opposite the crank, of sufficient size to balance, by its centrifugal force on the horizontal centers, the force of inertia of the reciprocating parts. If the counterweight is made heavy enough to do this, it will overbalance the vertical force of the connecting rod on the top and bottom points, and will, therefore, tend to make the shaft jump up and down in the bearings.

It is usually necessary to effect a compromise, having the weight of such size as to counterbalance the horizontal force in part, and yet not overbalance the vertical force too much. The foundation of the engine must take up the unbalanced force which cannot be handled by the counterweight.

Another thing which can be done to stop the reciprocating parts is to use a high compression at the ends of the stroke. This cuts down on the amount of power that the engine can deliver, but tends to prevent the

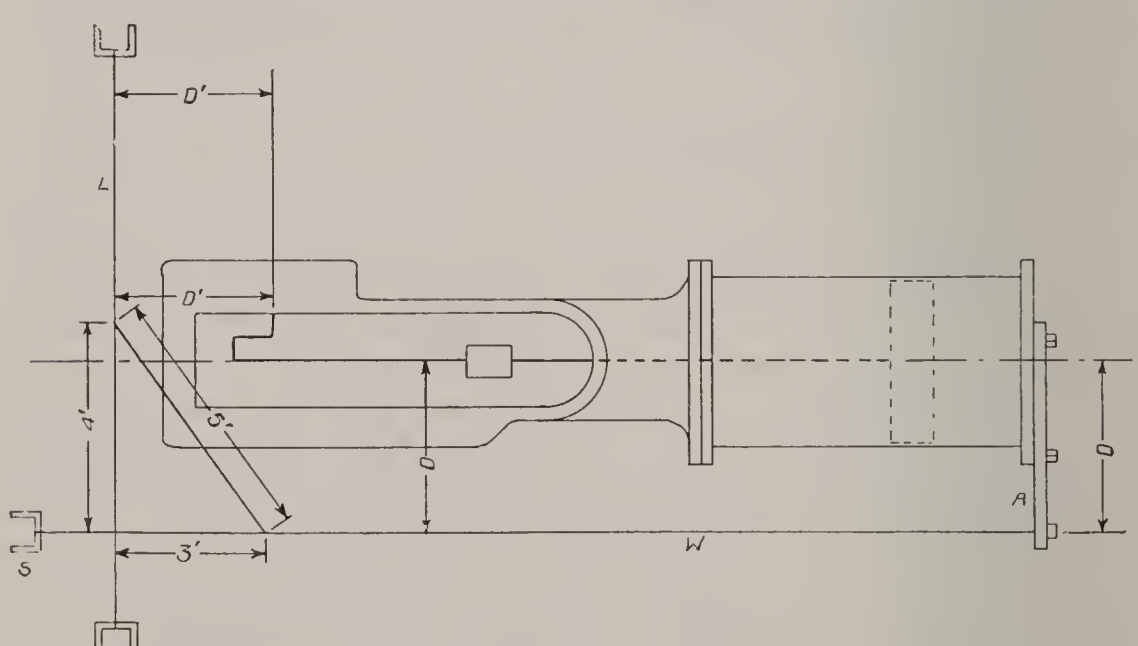
inertia shock on the shaft, transferring it, however, to the cylinder heads.

If the engine is mounted on a single cast-iron base, the vibration is due to a moving of this base on the foundation, or to the movement of the base and foundation together, and in this case, there is almost no possibility of doing anything with it except to put a big chunk of concrete around the whole base, making sure that it bonds well into the present foundation, and take up the vibration in that way.

There is no book on this subject that we know of, and such treatment of the balancing of engines as is found in books devoted to engine design and construction is rather complex and mathematical, intended more particularly for designers of engines.

401. Q.—How can I tell whether the shaft is in line with the cylinder?

A.—There are several ways in which the alinement of the shaft and cylinder may be checked. Nearly all



A401. METHOD OF SQUARING AN ENGINE SHAFT

modern engines have "spots" planed on the inside of the frame opposite the crank, and parallel to the center line of the cylinder. When this is the case it is only necessary to caliper between one "spot" and a point on the crank disk or end of the crank pin as most convenient, then turn the crank over until the same point on the crank pin comes opposite the other "spot" on the frame and caliper here. If the shaft is in line or

square with the cylinder, the two measurements will be the same. If not, one end of the shaft must be moved to bring them equal. If no "spots" are planed on the frame the shaft may be aligned in the following manner:

Take off the cylinder head, and bolt a strip of wood, A, to one of the studs so that the end will extend out past the cylinder, and about as high as the center of the cylinder. From this stretch a fine line or wire W to a support S beyond the crank shaft. This may be a post fastened to the floor, or a piece of wood fastened to the wall at the same height as the center of the cylinder. Now measure the distance D from the center of the counter-bore to the line at the end outside the cylinder, then lay off the same distance from the center of the piston rod and move the line up to it; then draw the line taut and fasten. This will make the line parallel to the center line of the cylinder. Then stretch a second line l parallel to the center line of the shaft and at the same height as the center, fastening it to supports as before. Square this line with the line first set, which may be done with a square or by the proportions 3, 4, 5. The latter is the best method, and is done as follows: Measure off 3 ft. on one line from where the two lines cross and tie a string on the line here. Then lay off 4 ft. on the other line and tie a string here; when the two lines are at right angles to each other the diagonal distance from one string to the other will be 5 ft. After the line is set correctly, measure the distance from the center of the crank shaft to the line on each end, and adjust the shaft accordingly. The accuracy of this method will depend on the accuracy of your measurements. If the piston, piston rod, and crosshead have been removed the line can be run through the cylinder but the line outside will do as well if care is taken to get the measurements correct.

402. Q.—How can I tell whether or not the engine shaft is at right angles to the center line of the cylinder without taking the engine apart?

A.—As the head of an engine cylinder is machined accurately, place a straight edge across the back of the head in such a manner that it is approximately level and at about the same height as the engine shaft. Support it well at intervals so that it will be free and not strained in any way.

See that it is snug up against the back head and then by means of a steel tape or other accurate means that the case at hand and the appliances available may suggest, measure the distance from the straight edge to the shaft, once as close to the main bearing as possible and again at the outboard bearing, making all due allowances for any difference in the diameter of the shaft which can be found by calipering.

If the engine is a center crank, proceed as above, only place the straight edge so that it projects far enough on both sides of the cylinder so that the measurements can be taken at the ends of the shaft, or if direct connected to a generator or other machine, then as far as possible from the main bearing, so as to secure a better chance to detect possible errors in the alinement.

This method is not as accurate as that of squaring the shaft to a line through the cylinder and guides, but if care is taken very good results can be secured where time does not permit running a line through the cylinder as in A401.

403. Q.—How would you check the crosshead for alinement?

A.—A bored guide is used by most engine manufacturers at the present time and as these guides are bored concentric with the bore of the cylinder, the crosshead can be centered by adjusting the shoes until the hub of the crosshead, if turned (or the piston rod just back of it if it is not), calipers the same all around from the bored guide. Engines having bar guides are not likely to get out of line, but in case of necessity, the best way is to remove the reciprocating parts, stretch a line through the cylinder and set the guide bars parallel to it and at such distance from each other so as to receive the crosshead when centrally located.

404. Q.—How would you determine whether or not the crosshead shoes were in alinement?

A.—Put the crank on the outer center and by means of the crosshead adjusting screws raise or lower the crosshead until the piston rod is in line with the lower guide, and then raise the crosshead from the guide enough to insert narrow strips of common writing paper under each of the four corners of the lower shoe. Drop the crosshead on the strips of paper and then try to pull

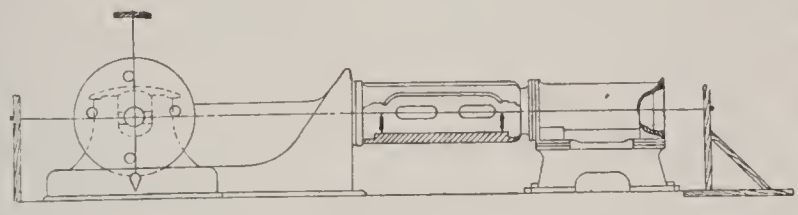
them out one at a time. If all are equally tight the shoe is in line with the guide, but if one or more pull out easily the face of the shoe is not in line with the guide and should be scraped until it is.

405. Q.—How should one proceed to make a complete alinement test on a steam engine?

A.—In the case of a new engine, first, make sure that the foundation bolts, or the bolt holes, check up with those in the engine frame. Then place the frame in position, with the bolts in place, and level by means of wedges, lengthwise for the lower guide and crosswise for the main bearing or the surface where the cap fits. The wedges should be placed close to foundation bolts, two under the bearing, one on each side at the front end of the guides and the same at the back end of guides, and the frame brought to exact level.

The cylinder may now be attached to the frame, making sure that there are no burs on the finished faces, and that it is level crosswise and lengthwise. Then draw up on the stud bolts equally all around several times, watching the levels all the time until the frame seems to be tight in place.

A line should then be stretched through the entire engine and fastened to adjustable targets, either on the frame and cylinder, or, better, at a distance behind the cylinder and in front of the main bearing, using a fine silk or sea grass line, stretched tight enough to take out the sag, and arranged in such a way as to permit of a slight adjustment in any direction.



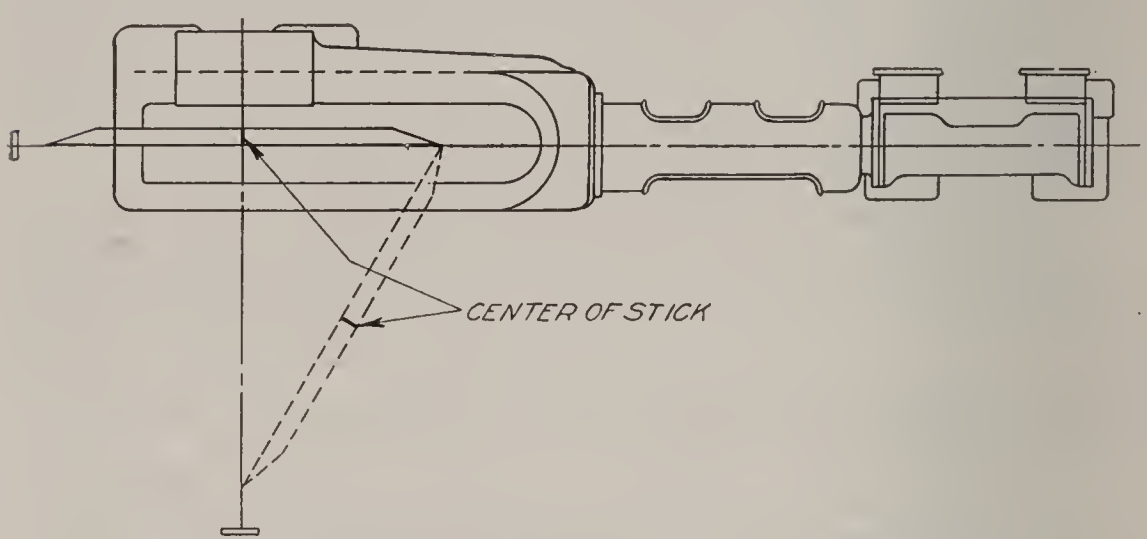
A405-1. CENTER LINE FOR CYLINDER AND GUIDES

Secure several small sticks of soft wood about the size of a lead pencil, cut them ½ in. shorter than the radius of the cylinder and of the guides, and in the end of each stick place a small brad rounded on the point. Then by tramming top and bottom and sidewise, adjust the line so that it is exactly central to the guides at each end. Now test the cylinder to see that it checks up with the line; finding the amount that it is

out if any by using a thickness gage or paper of known thickness under the tram.

Truing the Frame.—If the cylinder is out of line, and not over 0.012 in. low at the back end, this can usually be taken up by wedging up the rear end of the cylinder, watching to see that the level of the frame and adjustment of the line in the guides has not been affected. If, however, the cylinder is out sidewise, the only remedy is to back off the cylinder far enough so that the end of the frame, or the head, can be filed.

After the filing, it is, of course, necessary to bring all parts of the cylinder into place again, draw up on



A405-2. SQUARING THE SHAFT BY CENTER LINES

the stud bolts evenly, being sure that all parts are kept carefully in line and again test for the alinement.

For Bar or Slipper Guides.—If the engine has bar or slipper guides, it is best to set the stretched line to the cylinder first, and if it lines with the crankpin on both centers, to set the guides so that they will line with the axis of the cylinder and be at the right distance from the center line to take the crosshead. This usually involves the use of shims under the guide bars.

After cylinder and guides are properly lined, the main shaft must be squared up to the center line. It is best to have the shaft set so that its center will be in the same plane as the center line of the engine, but, if the shaft is a small amount high or low, it will not materially affect successful operation.

Squaring the Shaft.—Having brought the shaft level, adjust the outboard end sidewise so that when the crankpin is brought against the stretched line at either center, the distances from the line to the face

of the crank, or to the end of the pin, will be the same at both centers.

If it is desired to line up the bearings before the shaft is set into them, this may be done by stretching a line through the center of the main bearing to the outboard bearing, then moving the outboard bearing sidewise until this line is at right angles to the center line through the cylinder and guides.

To make sure that the lines are at right angles, a strip of wood 6 or 8 ft. long is pointed at each end like a lead pencil, and divided exactly at the middle. Hold this along the main center line with the middle mark at the point where the shaft line crosses, and mark the main center line at each end of the strip. Then hold one end of the strip at one mark, being careful not to push the line sidewise, and bring the other end of the strip to touch the shaft center line, marking the point. Next try the strip from the other mark on the main center line, and see if it touches the shaft center line at the same point as before. If it does, the lines are at right angles; but if not, the outboard bearing must be moved until this test is satisfactory.

The outboard bearing can then be raised or lowered by the adjustment provided by the makers, until a straight edge laid through the main bearing and the outboard bearing shows level.

After this last operation, it is desirable to verify the leveling and the lining up of guides and cylinder, the lining up of the main and outboard bearings and their leveling, to make sure that the work has been correct and that no alteration has been made at any point in making the adjustments.

If the lining up is done on an old engine, it may be necessary to rescrape the boxes when replacing the shaft. This can be done by coating the shaft with red lead, laying it in the bearings and revolving it to find where the high points come, and scraping these down until a satisfactory bearing is secured.

In the case of a cross-compound engine, not only must each side be in line with itself, but the two sides must be parallel, and both must be square with the shaft. This involves, of course, greater complication and care than in the case of the simple engine, but the method of operation is the same. In the case of a new engine, frequently the distance shown on the blueprints from

center to center of the two cylinders will not correspond to that in the actual engine, and one side or the other must be shifted the necessary amount to bring each cylinder in line with its crankpin.

Crank and Crankpin.—To test the squaring up of crankpin and crank on the shaft, revolve the shaft until the crankpin touches the line through the cylinder and guides, and measure the distance from the face of the crank to the line. Revolve the shaft until the crankpin touches on the other side, and measure the same distance. Then drop a plumb line through the center of the shaft, just touching the main center line, Fig. A405-1, revolve the crankpin until it touches this plumb line, measure the distance from the line to the crank, revolve the pin to the other vertical position and again measure this distance. The two horizontal measurements and the two vertical measurements should all be equal.

Piston and Crosshead.—The piston and crosshead may now be put in, the piston centered in the cylinder by the junk ring and the distance of the piston rod center from the lower guide be measured near the stuffing box. The crosshead shoe should then be adjusted until the center of the crosshead is at the same distance from the lower guide, when the piston rod may be connected to the crosshead.

Crosshead Pin.—The connecting rod may be attached to the wrist pin and keyed up snug, then placing the crank on center, push the crosshead over until the end of the connecting rod rests on top of the crankpin. If the wristpin is in proper line, the end of the connecting rod will be at the same distance from both ends of the crankpin.

Eccentric.—To line up the eccentric, fasten a line at the middle of the eccentric rod pin, and, holding a square against the shaft, press the line against the other leg of the square near the shaft and move both square and line until the line touches the square throughout its length. Mark the point where the line touches the shaft and on each side of this mark at distances equal to half the thickness of the eccentric make other marks and shift the eccentric to a position between these.

Connect up the engine, being sure that all brasses fit snugly on their pins with a smooth fit. Give the

bearings plenty of oil and start up carefully, watching all bearings for any signs of heating. There may be a slight warming up at some points, which will soon disappear, but if any serious heating occurs, the bearing should be tested up carefully to find out where the difficulty lies.

406. Q.—To USE exhaust steam for heating, what back pressure should be carried on an 18-hp. slide-valve engine? Pressure carried is 5 lb., but would like to increase it if advisable. Boiler pressure is 80 lb.

A.—We do not believe that the back pressure on your engine could be increased appreciably without a decided loss of power. Ordinarily, the loss of power is nearly proportional to the ratio of the back pressure to the mean effective pressure.

Of course, there are two ways of offsetting this loss; one by raising the initial pressure and the other by increasing the cutoff. Engines are usually designed to work most economically at a given cutoff, so that in most cases it is undesirable to change it to any extent.

Computations for an engine having a clearance of 5 per cent, using 80-lb. steam and cutting off at ¼ stroke, show that raising the back pressure 3 lb. calls for a corresponding increase of 5 lb. in the boiler pressure to maintain the same engine horsepower.

As we believe a pressure of 5 lb. should prove ample to circulate steam through any well-designed heating system, even of comparatively large size, the increase of back pressure calling for a higher boiler pressure scarcely appears justified.

407. Q.—To RUN one end of an engine exhausting to a heating system at 5 lb. back pressure, and the other end condensing, which end should be connected to the condenser, and which end would do the more work?

A.—If one end of an engine is to be run condensing and the other end exhausts to the heating system, with 5 lb. back pressure, it would be better to operate the head end condensing, on account of the air leakage around the piston rod. The condensing end would do the greater amount of work, unless the cutoffs were made unequal.

408. Q.—STEAM HEATING requires 5 lb. gage to be

taken on the exhaust of a compound non-condensing engine. Should the steam be taken from the receiver or from low-pressure exhaust?

A.—With the engine running noncondensing the steam from the heating system should be taken from the low-pressure cylinder exhaust pipe unless there is some other use for the exhaust steam. When running in this manner the benefits of compounding would be gained in the engine from the resulting lower initial condensation and re-evaporation, and although there would be a greater amount of back pressure on the low-pressure cylinder, all the heat in the exhaust would be available for heating. It would be advisable to have a return pump on the end of the returns so that at a time when a pressure of less than 5 lb. would supply sufficient heat the back pressure on the low-pressure cylinder could be reduced, and still have good circulation in the heating system. Whether taking the steam from the receiver or low-pressure exhaust, the extra amount of steam would have to pass through the high-pressure cylinder. If a compound engine is installed with the idea in view of taking the steam for heating from the low-pressure cylinder at 5 lb. pressure the ratio of the cylinders would be smaller, that is they would be nearer the same size than when running condensing. If the engine is to be run condensing when steam is not required for heating it might be better to take the steam from the receiver, or at least have a connection to both the receiver and the low-pressure exhaust pipe.

409. Q.—NAME EIGHT losses which tend to lower the efficiency of a steam engine.

A.—(a) Unavoidable thermodynamic loss; (b) im perfection of cycle employed; (c) imperfection of condensing machinery employed; (d) wire drawing and fluid friction; (e) cylinder condensation; (f) valve and piston leakage; (g) conduction and radiation of heat (h) mechanical friction.

410. Q.—TO WHAT cause may excessive back pressure on an engine be due? Is excessive back pressure objectionable and why?

A.—Excessive back pressure on an engine may be due to the fact that the engine is exhausting into a steam

heating system, or that the exhaust steam is being used in some manufacturing process system which would have a tendency to cause a decided back pressure on the engine. If, however, the engine is exhausting into the atmosphere, it may be possible that the line leading from the engine to the atmosphere is in some way obstructed. This may be brought about by too long an exhaust pipe, or the use of too many elbows.

Excessive back pressure is objectionable, as it has a tendency to reduce the amount of power which the engine is capable of developing, and to reduce heat efficiency.

411. Q.—What effect would unequal piston clearances have on the operation of a steam engine?

A.—Unless the piston on the engine actually touches either cylinder head, unequal clearances will have no dangerous effect on the operation of the engine. The end having the less clearance will have greater expansion and a lower terminal pressure, cutoffs being equal, and will develop less power. Slight inequality of clearances is, however, of no consequence.

412. Q.—What is the best remedy for a reciprocating engine that does not work well with superheated steam? No other steam is available.

A.—The best way to reduce superheat locally is by injection of water into the steam pipe at some distance before the steam enters the engine. The spray should be regulated so as to give as little water as possible and still get satisfactory action of the engine. The less water used, the better will be the efficiency of the engine.

Another remedy might be a forced lubrication system delivering oil direct to the bearing surfaces of valves, piston and rods.

413. Q.—Using a good grade of cylinder oil in large quantities and not obtaining proper cylinder lubrication, where would you look for the trouble?

A.—If a cylinder is not being lubricated properly with a good grade of oil, look first to see that the oil gets into the cylinder properly—that is, is not carried away by gathering on drops of water in the steam and being thrown out by the separator before the steam gets to the engine. (Oil should be introduced between the separator and the engine.) If this is not the trouble,

test to see whether the steam goes into the cylinder wet or superheated.

In the case of superheated steam, it is much more difficult to get the oil to cling to the cylinder walls. It is apt to diffuse through the steam and but little of it to reach the surfaces which need lubricating. Admixture of some animal oil may be helpful.

414. Q.—How MANY drops of cylinder oil should be fed into the cylinder of an 11 by 16-in. engine running 205 r.p.m.? Steam pressure 100 lb. gage, saturated. The engine is an old one.

A.—On an engine of the size and under conditions which you state we would suggest using about 3 drops of cylinder oil per minute. Use the least amount you can without having the engine show signs of distress.

415. Q.—How MUCH heavy-body cylinder oil should a 16 by 36-in. Corliss engine running at 120 r.p.m. and using saturated steam at 100 lb. pressure, use in 12 hr.?

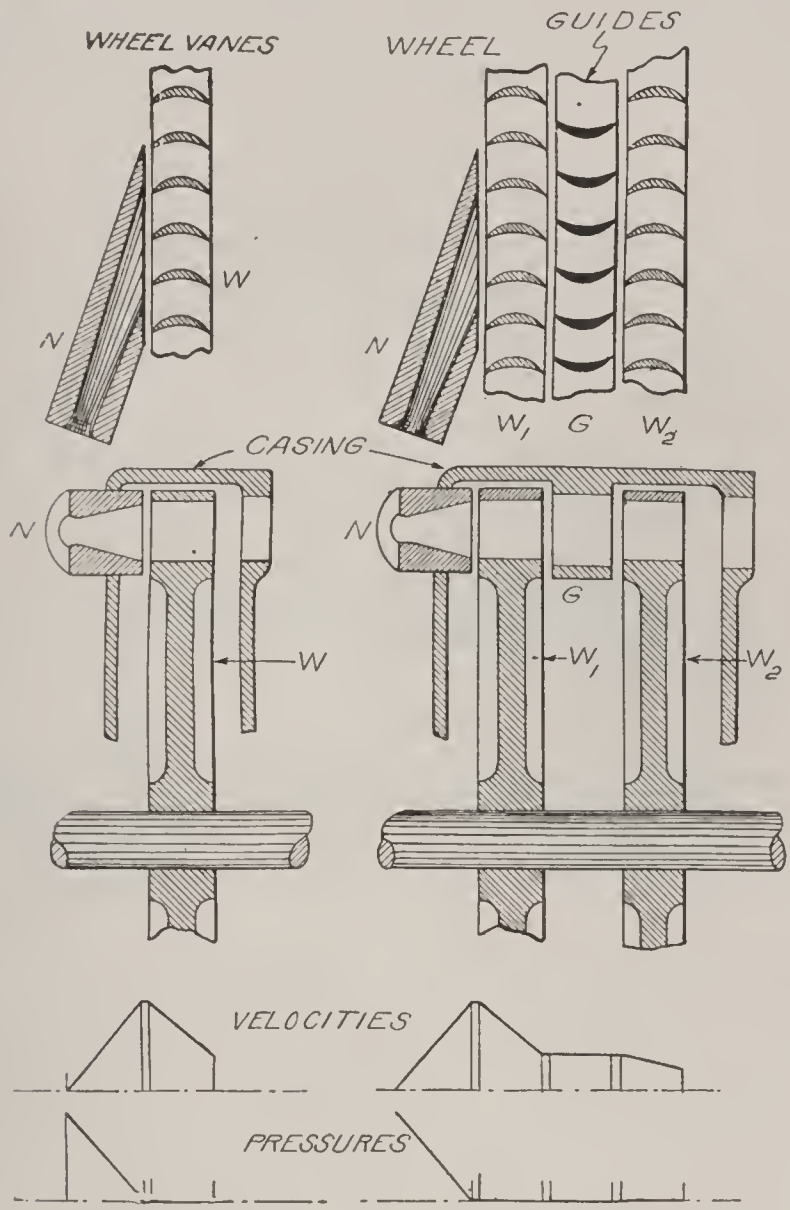
A.—The subject of cylinder lubrication is open to extensive discussion and no fixed rate of feed will cover all engines. The writer heard of one 16 by 36 Corliss engine, running at 86 r.p.m. with 120 lb. steam, which was successfully operated with one drop for every 6 revolutions. Operating at 120 r.p.m. with 100 lb. steam, would, on the same basis, probably require about one drop for every 5 revolutions. Cylinder oil may contain from 4000 to 6000 drops per quart, thus it is difficult to estimate the 12-hr. consumption in liquid measure without knowing the viscosity of the oil.

To determine the proper rate of oil feeding, it is recommended that you cut down the supply until a slight tremor is noticed in the eccentric rods or, possibly, in a Corliss engine, a decided groaning is detected in one or more of the valves. If the oil is increased sufficiently to give the valves good operation again, after repeated trials, the engineer can determine how much of a particular brand of oil is required to keep the valves working smoothly and quietly.

The next thing in order would be to take out one of the valves and examine the wearing surfaces; if well lubricated it will present a dark, glossy appearance and enough oil to show on a piece of white paper when rubbed over the surface.

416. Q.—IN A Curtis turbine, the pressure being the same all around the bucket wheel, what effect does this have on the motion of the steam turbine?

A.—In the Curtis turbine, the heat energy stored up in the steam is imparted to the wheel or wheels, both



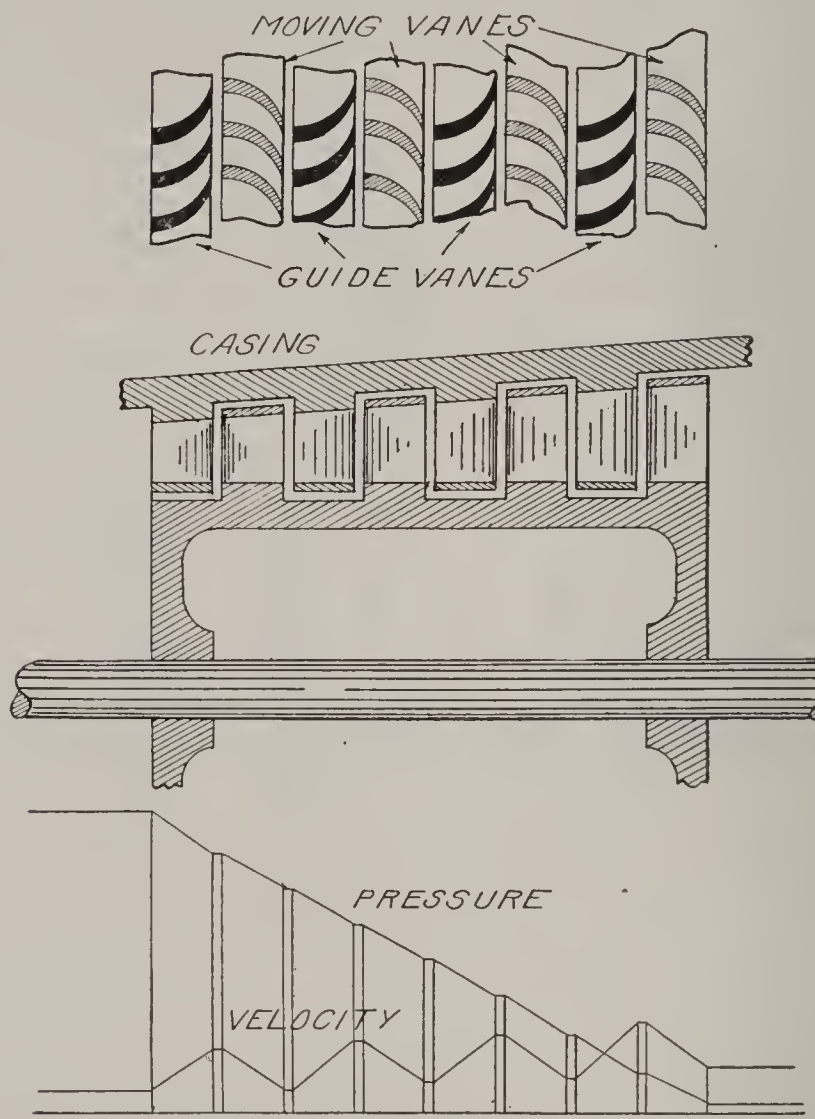
A416. DIAGRAM OF CURTIS NOZZLES AND WHEELS

by impulse and reaction. The steam passes through expanding nozzles in which most of the expansive force is transformed into the force of velocity. The steam passes through one or more stages of the revolving element, or wheels, depending on design, each stage or wheel having its own set of expanding nozzles and each succeeding set of nozzles being greater, both as to number and area than the preceding set. The ratio of expansion in the several sets of nozzles depends on the

number of stages, much the same as in a reciprocating engine with several stages or cylinders.

417. Q.—How does the steam cause the wheels to rotate in a Westinghouse turbine?

A.—The Westinghouse turbine is of the reaction type and consists of two principal parts, the rotor and the casing. The rotor consists of a shaft carrying steel drums on which are mounted rows of blades.



A417. DIAGRAM OF WESTINGHOUSE-PARSONS GUIDE AND ROTOR BLADES

Those at the entrance end are quite small, possibly not over ¾ in. long and in each successive row the blades are larger than in the preceding one until at the exhaust end they are several inches long.

The casing has the stationary blades attached to it in such manner that they are opposite those on the rotor, those on the latter rotating between the rows of the former. The steam enters where the casing is

smallest where are located the smallest blades. Immediately the steam begins to expand in passing successively from one set of blades or stage, to another, striking in its passage the curved surfaces of both the rotor and casing blades, reacting from one to the other, thus causing the rotor to revolve.

At each successive set of blades, the pressure is reduced until at the exhaust nozzle where the steam enters the condenser its pressure is equal to about 1 or more inches of mercury, and of a corresponding temperature.

418. Q.—How would you start up an Allis-Chalmers turbine?

A.—Get the oil pump going until it lifts the inlet valve, develops about 20 lb. on relay oil gage and 2 to 4 lb. on bearing gage. Warm up by opening throttle for ½ hr., then run slowly up to speed. Turn water on glands.

419. Q.—How do you stop a turbine?

A.—Start up the auxiliary oil pump to hold the oil pressure as the turbine slows down, break vacuum when the load is nearly off, and otherwise reverse the order of starting.

420. Q.—What is meant by the critical speed of a steam turbine?

A.—The critical speed of a steam turbine is the speed at which the vibration is the greatest. In starting up a turbine for the first time if possible to do so, the speed is watched closely and if there is any vibration at certain speeds, these speeds and the intensity of the vibrations are noted.

This critical speed is not always uniform in machines of the same size. The writer is familiar with two turbines identical in every particular, so far as human skill can make them, both running at 3600 r.p.m. On one, the critical speed is at about 3500 and on the other, hardly noticeable at about 3300 r.p.m.

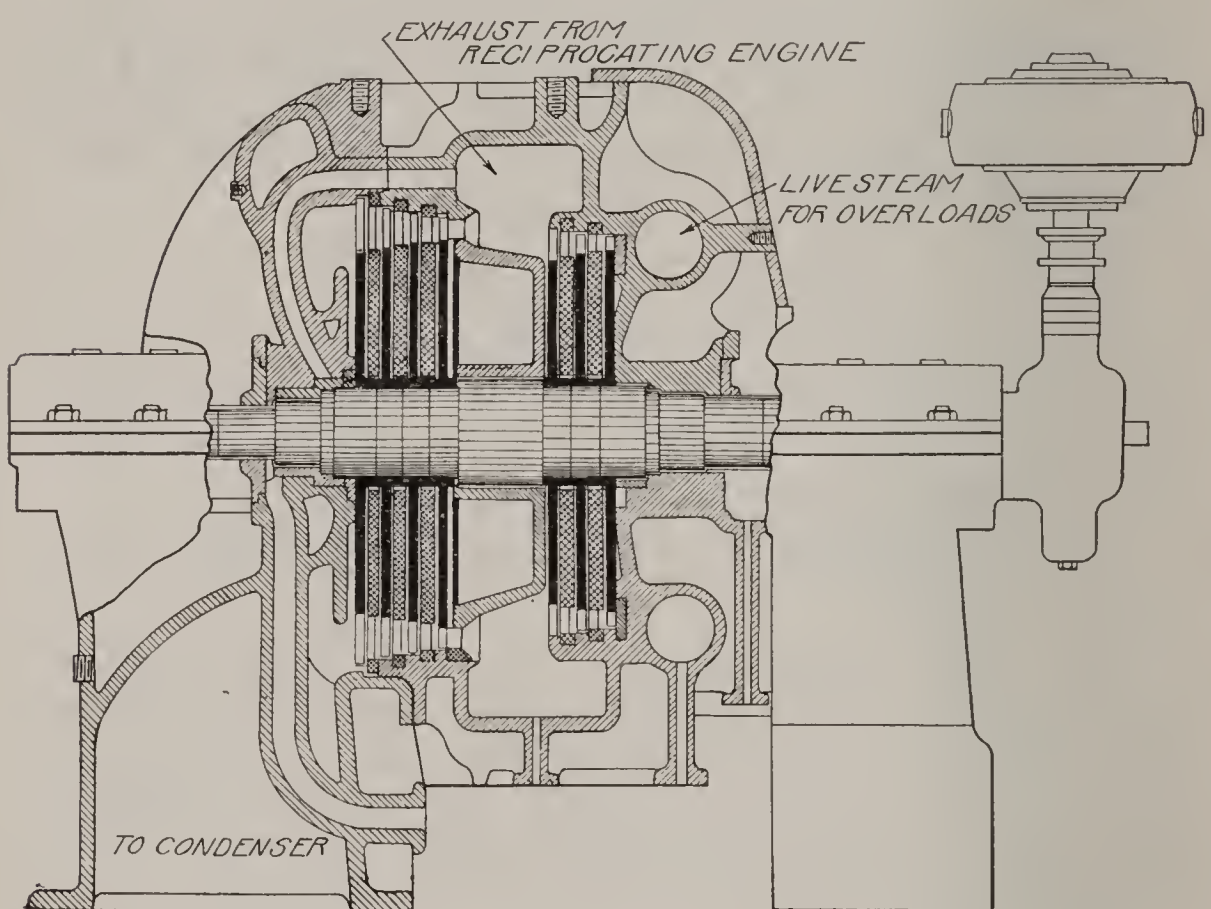
Of course it is not best in bringing the machine to speed to let it run long at critical speed but hasten it a little with the throttle to pass the point. The same phenomena is also noted with wood planers, high speed generators, and other rotative machines operating at comparatively high speeds, and it sometimes happens

that the critical speed is identical with the operating speed.

In case of a planer or high speed generator, this vibration can be eliminated by balancing. This critical speed of machines is frequently aggravated by sympathetic vibration in some part of the machine itself or of the foundation, especially if it rests on a floor or even on a substantial foundation on poor soil.

421. Q.—What is a mixed-pressure turbine?

A.—A low-pressure turbine with a section fitted with special expanding nozzles to receive high-pressure live



A421. RATEAU MIXED FLOW TURBINE

steam from the boiler; this is better than using a reducing valve. This extra supply is governed by the regular governor.

422. Q.—What is a double flow turbine?

A.—One in which the steam enters at the center and flows towards both ends; this takes care of the end thrust and does away with the end thrust balancing pistons.

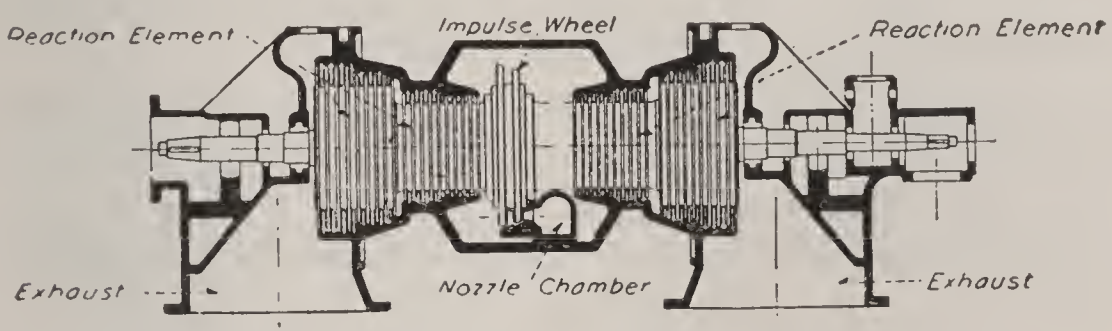
423. Q.—What is the pitch of blades?

A.—The distance between the edge of one blade and the edge of the one next to it.

424. Q.—What keeps a steam turbine rotor from going endwise?

A.—End thrust in steam turbines is usually avoided by the use of balance pistons, or by making provisions for the steam to flow in opposite directions from the center.

425. Q.—Why is the larger type of Allis-Chalmers turbine made with a larger and shorter rotor? Is the



A422. WESTINGHOUSE DOUBLE FLOW TURBINE

velocity of steam in a reaction turbine higher where it enters the turbine or where it leaves the turbine and enters the condenser?

A.—The larger type of Allis-Chalmers turbine is fitted with an impulse wheel which does the work equivalent to that of the first two or three stages, thus shortening the length of the rotor. The comparatively large diameter of this impulse wheel gives a turbine with this combined impulse and reaction arrangement, a larger and shorter appearance. In a reaction turbine, the steam velocity is greater at its exit.

426. Q.—What is the vertical bearing used for on a Curtis vertical steam turbine.

A.—In the case of vertical Curtis turbines, it is intended to have the shaft always run on an oil or water film between the bearing surfaces, as no metal would long stand actual contact under such severe conditions. The oil or water is pumped into the bearing under considerable pressure, which in the case of 5000 kw. units runs up to 1000 lb. per square inch. Under this pressure, the fluid is forced between the rubbing sur-

faces of the bearing and keeps them from coming into metallic contact with each other. The thickness of the interposed film of oil is about 0.003 in.

The surfaces of the bearing are of cast iron, since this material wears quite uniformly, even under actual metallic contact. Hardened steel and brass have been tried, but have been abandoned in favor of cast iron, because they cut badly when metallic contact occurs.

The oil or water is supplied by means of high pressure pumps, of which there are always two or more, so that should one be disabled, operation of the turbine will not be interfered with.

427. Q.—If the supply of lubricant for the step bearing fails, what will be the result?

A.—The step bearing will probably weld together, so that it would be necessary to have new bearings. Gradual wearing will cause the bucket wheels to rub on the stationary blades, and the friction will stop the machine.

428. Q.—What causes the governor of a 75-kw. Curtis turbine to stick and fail to admit steam when first being started?

We are compelled to lift a lever before the turbine will take steam. After starting, however, it works satisfactorily.

A.—We should judge this to be a 75-kw. Curtis steam turbine with mechanical operated valve gear which consists of a number of separate valves which are opened and closed by dogs fastened to a reciprocating lever rod. The dogs have a continuous reciprocating motion and whether or not they operate the valve depends upon the position of a series of shield plates which is operated by the governor. When a valve is once opened and turbine is in operation, pressure of steam in the bottom of valve keeps it open.

In addition to the above, the packing around the valve stem should be tight enough to prevent the valve, which is very light, from seating. When the throttle valve is closed, and the machine comes to rest, the governor causes all valves to be opened and they should remain in this position, due to the friction of the packing on the valve stems. It sometimes happens that packing

is worn and does not hug the valve stems tight enough to keep them up.

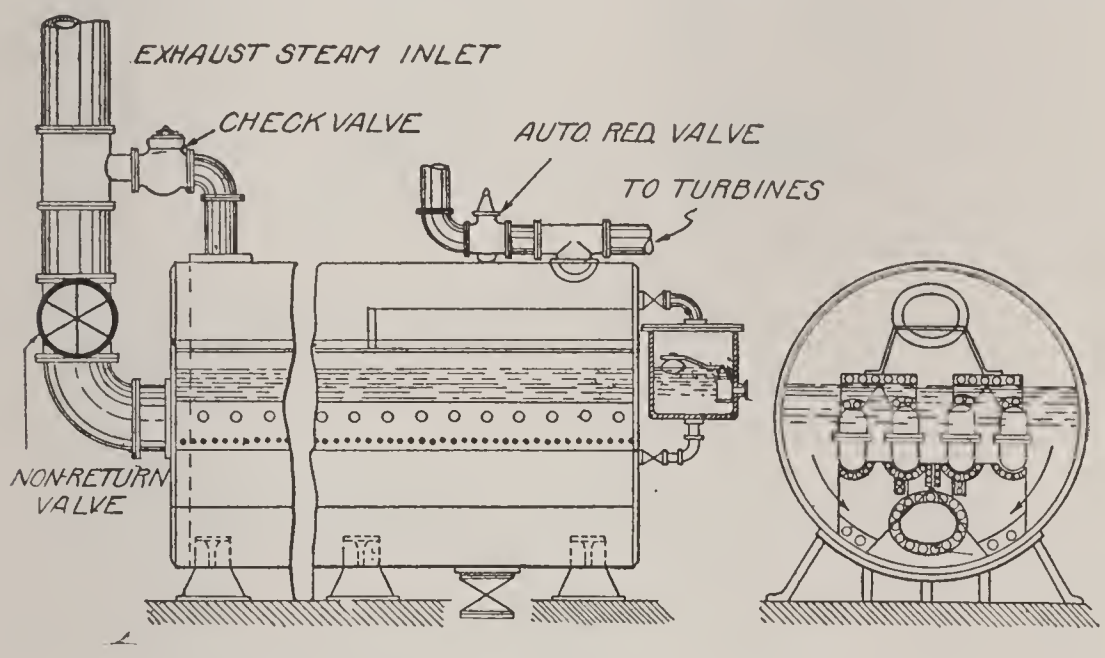
We believe if you will screw down on the stuffing-box controlling the valve stem packing, that the difficulty will be eliminated.

429. Q.—Why can you safely start an impulse turbine quicker than a reaction type?

A.—The clearance is so large in the impulse type that the throttle may be opened as fast as you can open it.

430. Q.—Describe the construction and operation of a regenerator or heat accumulator as used on a turbine.

A.—An accumulator is not an integral part of a turbine but an apparatus placed between an engine and a low-pressure turbine. It is usually a cylindrical

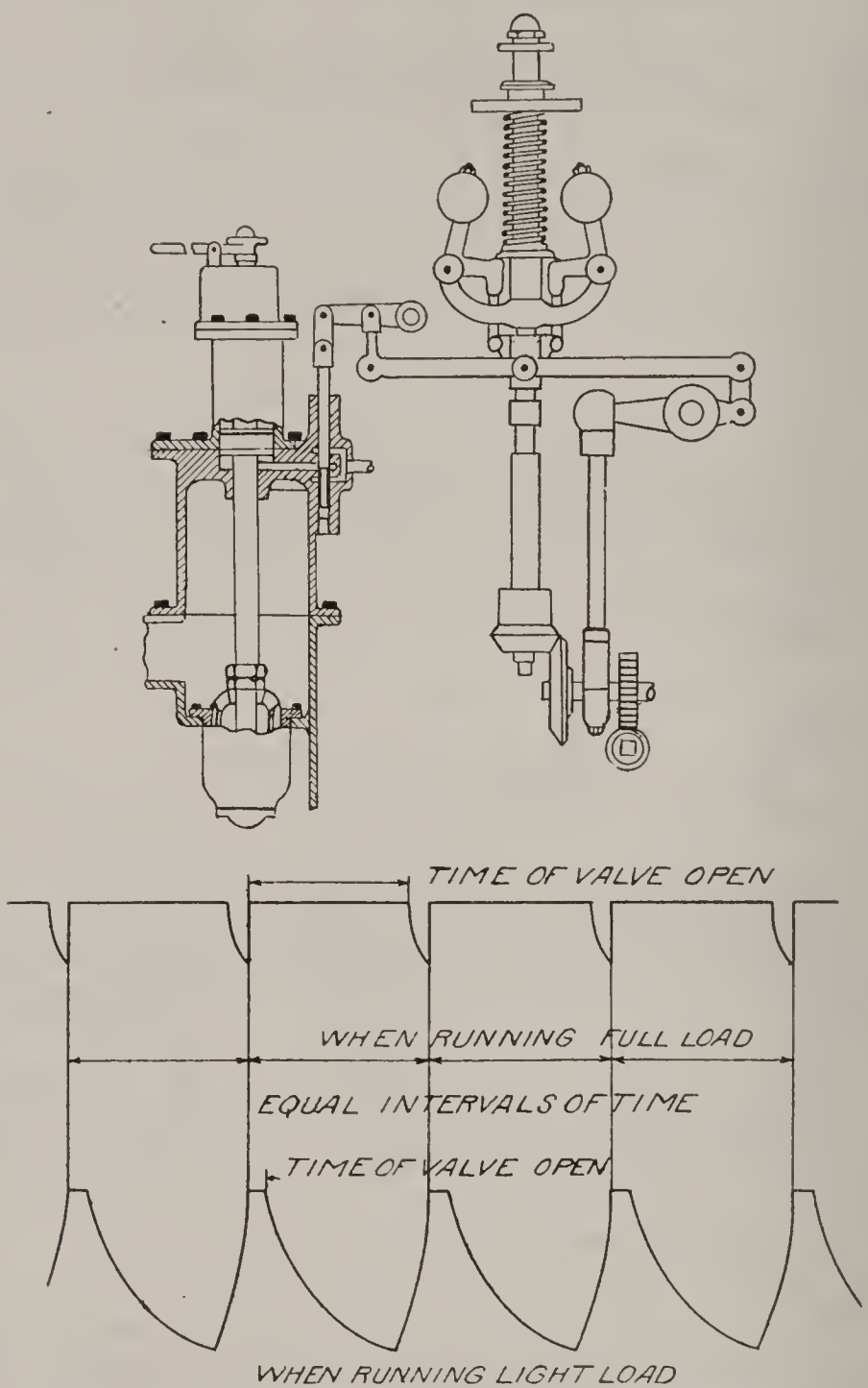


A430. Rateau steam accumulator

tank containing water which is kept in forced circulation mechanically or by the injection of steam. Steam collecting in the accumulator is condensed as it arrives from the engine in larger quantities than the turbine can use, and is again vaporized during the interval when the exhaust of the engine diminishes. The necessary variations for condensation and regeneration of the steam correspond to fluctuations in pressure in the accumulator, this pressure increasing when the apparatus is taking up heat and decreasing when it is discharging into the turbine.

431. Q.—How does the governor of a Westinghouse turbine work?

A.—Admission is through a double beat valve controlled by a steam operated piston above it, the valve



A431. WESTINGHOUSE GOVERNOR AND DIAGRAM OF STEAM CONTROL

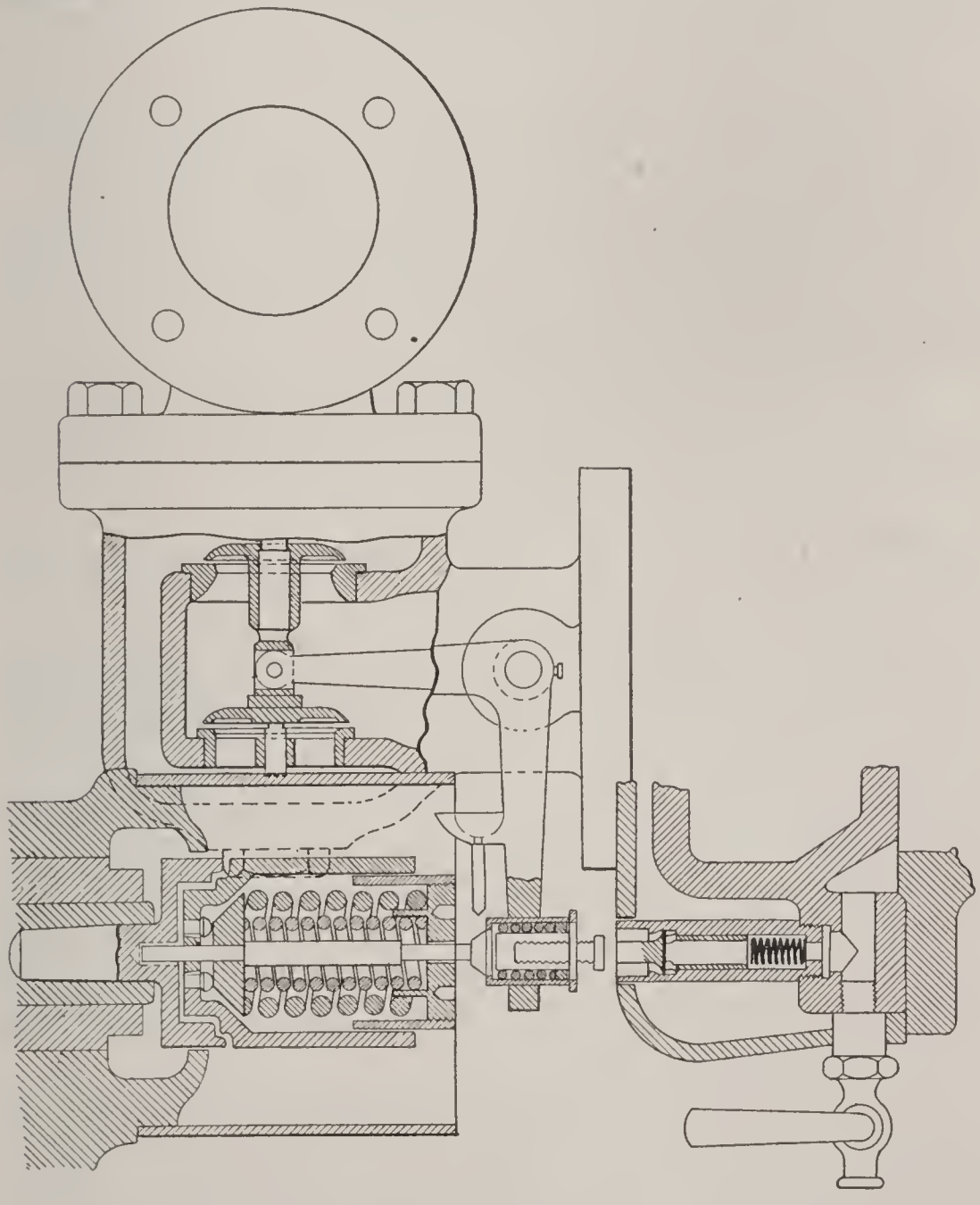
leading to the piston being operated by the governor. A heavy spring on the piston holds the main admission valve to its seat. The governor gets its motion through spiral gear and eccentric from main shaft. The valve is continually vibrating so that it is easily moved by the governor, which controls its mean position.

432. Q.—On the Allis-Chalmers turbine how does the governor work?

A.—This governor is of the Hartung type. It is driven by cut gears from the shaft working in an oil bath; this operates the balanced oil relay valve only. The two steam valves, main and bypass, are controlled by oil pressure of about 20 lb acting on a piston.

433. Q.—How is the Curtis low-pressure turbine governed?

A.—By varying the pressure instead of governing by nozzles; they may have hand operated valves in the first stage which accomplishes the same results as to light loads and economy as does the governor on a high-pressure turbine.



A434. DE LAVAL THROTTLING GOVERNOR

434. Q.—How does the throttling governor on a DeLaval work?

A.—The governor has a spring-actuated double beat valve, controlled by spring and weight governor at one end of the gear shaft reduced in speed 10 to 1. The governor also operates a tappet which comes into contact with an independent relay valve that operates to close the main throttle valve.

435. Q.—What will prevent a turbine from running away with no load?

A.—There is a safety stop mechanism which trips the main valve or an auxiliary stop valve. When running, the valve is held open by a latch; when the turbine runs above normal speed, the speed governor works and trips the latch and the valve closes.

436. Q.—Is there any set rule by which to calculate the amount of steam consumed in a turbine? Also, how do you find the horsepower of a steam turbine?

A.—There is no rule for finding the steam consumption in a turbine, or for finding the horsepower. The horsepower must be determined by a brake test, and the steam consumption by a steam test.

The only method is to weigh either the water fed to the boiler which is supplying the turbine or the condensation from the condenser.

437. Q.—How is the horsepower of a turbine figured?

A.—In the majority of cases, turbines are direct connected to generators, and the power is most readily measured by the instruments on the switchboard, which readily show the output of the generator, instead of the power developed by the turbine. Having the output of the generator in kilowatts, and neglecting any generator losses, such as copper losses, hysteresis losses and windage losses, you can readily determine the horsepower developed by the turbine by multiplying the number of kilowatts indicated by the instruments on the switchboard by 1.34.

The other method used to determine the horsepower of a turbine would be to apply a Prony brake, and by means of this determine the brake horsepower. Due to the fact that there has not been, up to the present time,

any indicator devised for the purpose of determining the steam horsepower, it is impossible to indicate a steam turbine.

438. Q.—On reciprocating engines, as a rule 25 to 26 in. vacuum is carried—on turbine 28 in. or more. Why is the extra vacuum used with turbines?

A.—With steam turbines, the advantage gained by reduction of back pressure is more marked than with the reciprocating engine, though theoretically the same for the same range of expansion. Initial condensation, leakage past valves and other sources of loss prevent a reciprocating engine from benefiting from a good vacuum to the same extent as a turbine, and the cost of the larger cylinder to take advantage of high vacuum is prohibitive.

PART III

Auxiliary Steam Equipment

Condensers

439. Q.—How does a condenser increase the horsepower of an engine?

A.—A condenser is usually used in connection with a steam engine, for the express purpose of obtaining more power out of the engine with a given amount of steam. By examining the indicator card of an engine operating noncondensing, you will find that the lower end of this card does not touch the atmospheric line, but will generally be somewhat above this, depending on the back pressure on the engine.

Now, by employing a condenser, the lower end of the card will be found considerably below the atmospheric line, or, in other words, the area of the card has been increased, as has also the mean effective pressure.

As the horsepower of an engine is directly proportional to the mean effective pressure you will have an increased horsepower brought about by this increase of card area.

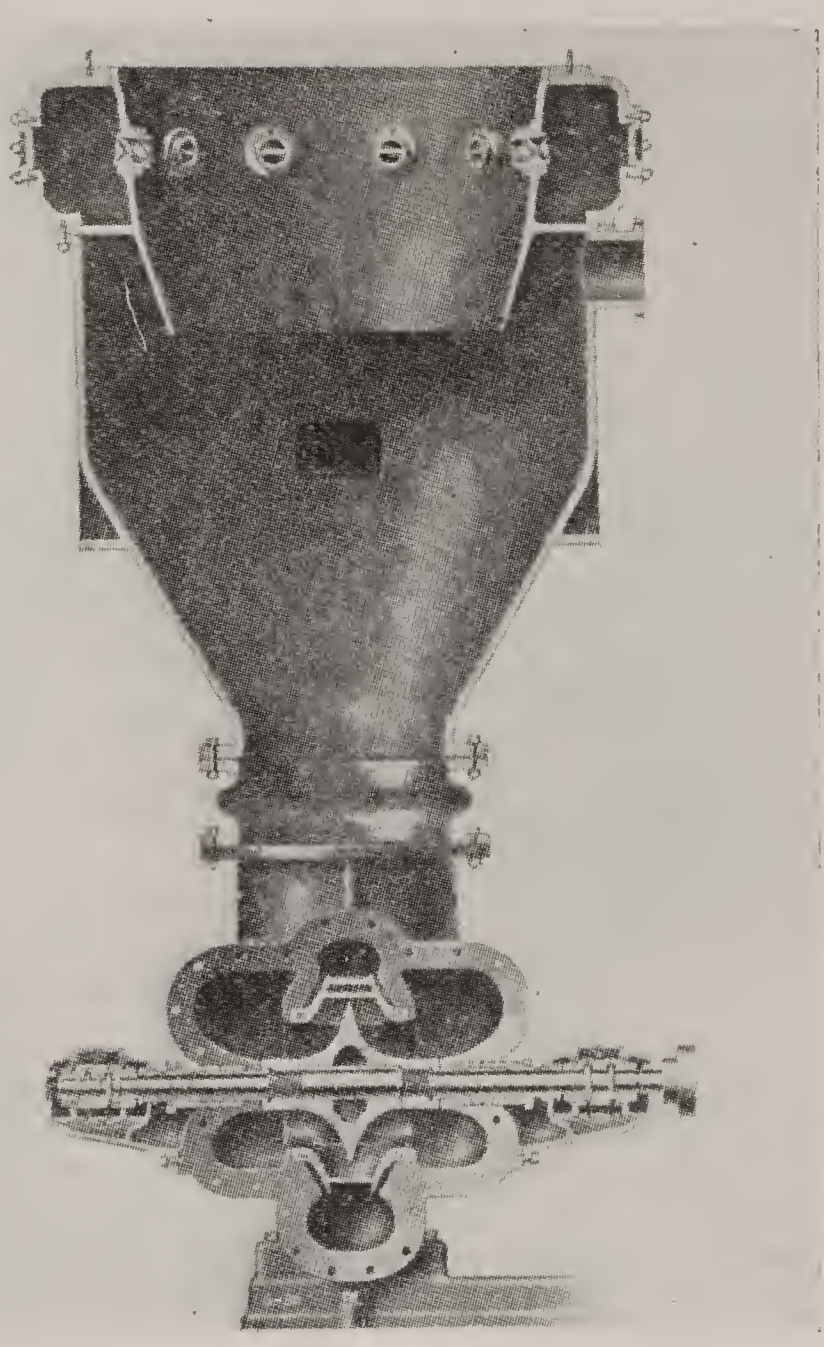
440. Q.—How do jet and surface condensers operate?

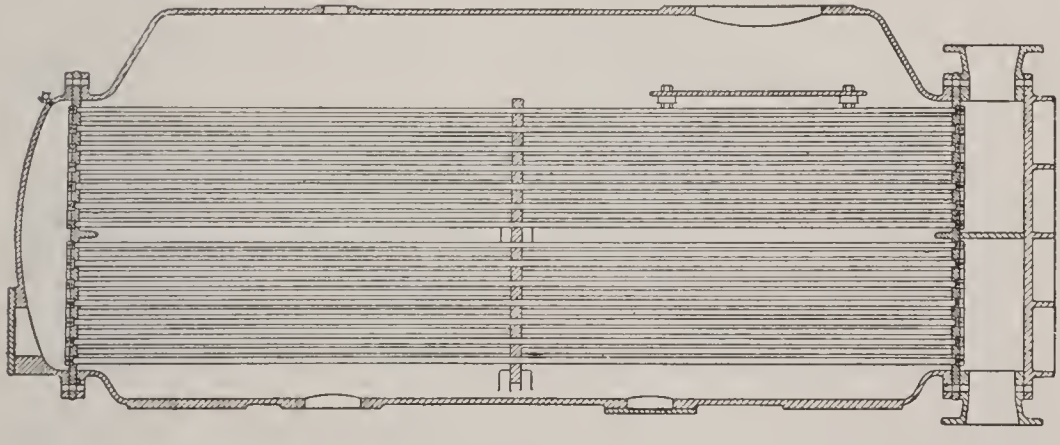
A.—A jet condenser is one in which the steam exhausted from engines or turbines is brought into intimate contact with a supply of water, which condenses this steam and creates a vacuum. A surface condenser is one consisting of a large chamber containing many small tubes, usually made of brass. Through these tubes, cold water is circulated, while around these tubes, the steam from the engines, turbines or pumps, as the case may be, is carried so that, as the steam comes in contact with these comparatively cold tubes, it is condensed, forming water and creating a vacuum.

441. Q.—What is the difference between a siphon and a barometric condenser?

A.—A siphon and a barometric condenser are practically the same thing. In both of them, the suction is secured by the use of a long tail pipe, extending downward from the condenser head, which is used to create the vacuum. Sometimes this suction is used to pull up

the cooling water from a lower level, sometimes the water is supplied to the condenser head by a pump, but in any case, the cooling water and the condensed

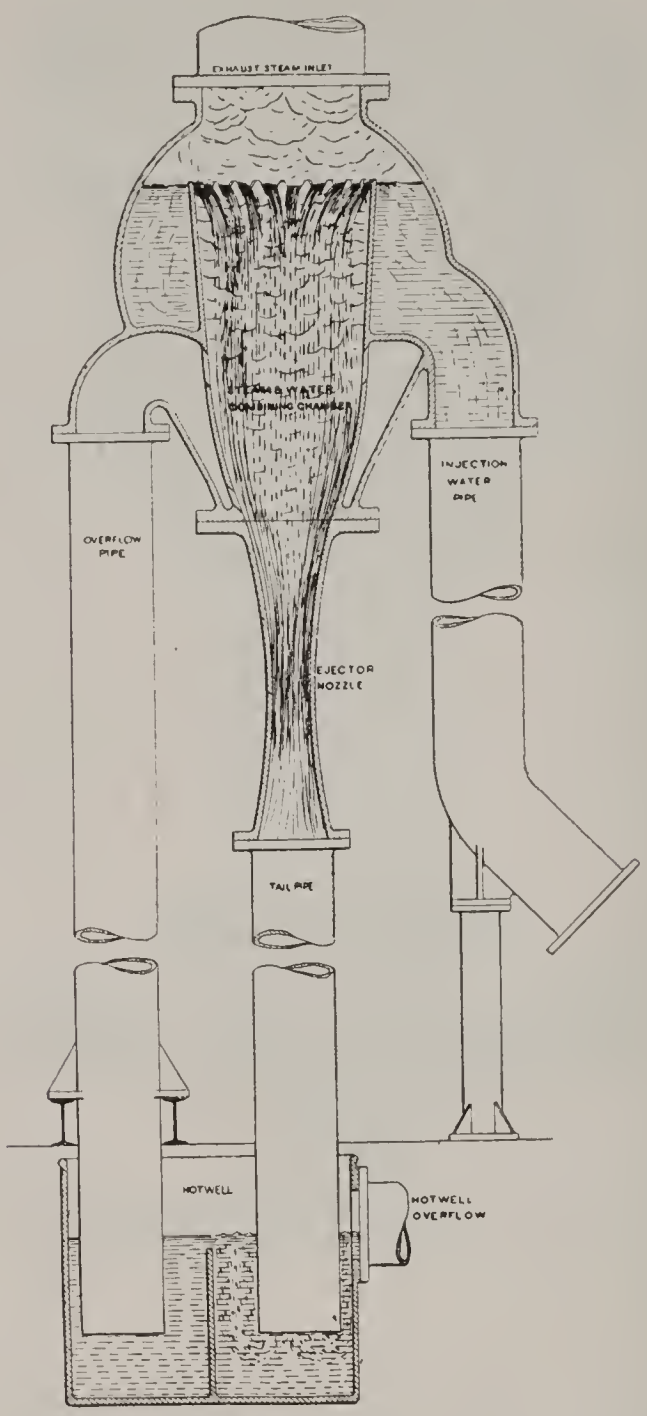




A440-1. TYPICAL JET CONDENSER. 2. TYPICAL SURFACE CONDENSER

steam are carried away by the suction of the falling column of water on the tail pipe.

In some recent types, the suction has been produced by pumps attached to the end of a short tail pipe, in-



A441. TYPICAL BAROMETRIC CONDENSER

stead of using a pipe long enough to give the suction by the falling of a column of water, but this, then, becomes really an ejector type of condenser, which has no particular advantage over the ordinary jet condenser.

442. Q.—In a siphon or barometric condenser why is it necessary to have the hot well 34 ft. below the condenser head?

A.—In a siphon or barometric condenser, the hot well should be 34 ft. below the condenser, so as to overcome the atmospheric pressure, and thereby prevent the drawing of water into the engine cylinder. Theoretically, the pressure of the atmosphere is equal to the pressure exerted by a column of water 34 ft. in height. A perfect vacuum is never reached, but it is well to allow a distance of 34 ft. in order to be safe.

443. Q.—WHAT IS the possible obtainable vacuum on a surface condenser when the outside barometric reading is 28.5 in.? Does the number of inches of vacuum decrease as the outside air pressure decreases?

A.—The vacuum obtainable depends, of course, on the effectiveness of the condenser as well as on the barometric pressure; but the obtainable vacuum decreases as the barometric pressure decreases.

Obviously, where your barometric pressure is 28.5 in., it would be impossible to get a 29-in. vacuum, although this is not infrequently done with a 30-in. barometer. Ordinarily, 1 in. less than barometric pressure is as well as can be done; although in some instances where the units installed are large enough to make it worth while, a vacuum within ½ in. of barometric pressure has been secured.

444. Q.—I AM getting 27½ in. of vacuum on a surface condenser and wonder if that is as much as I can get at this altitude of 1100 ft.

A.—The barometric pressure varies, as you are aware, with the weather, through a small range, and it varies also with the altitude, but not exactly in direct proportion. For instance, with a barometer of 30 in. at sea level, 29 in. would be 890 ft., 28 in., 1812; 27 in., 2767; 26 in., 3758; 25 in., 4787; 24 in., 5859; 23 in., 6977; 22 in., 8144; 21 in., 9366; 20 in., 10,648.

Your pressure of 28.5 corresponds to an altitude of 1347 ft., at a temperature of 32 deg. F. and 27.5 in. is as good vacuum as you can expect.

445. Q.—WITH BAROMETER at 30 in. and vacuum of 22 in., what is the pressure in the condenser?

A.—At 62 deg. F., 30 in. of mercury is accepted as equivalent to 14.7 lb. per square inch, so that 22 in. will

be equivalent to 10.48 lb.; 14.7 — 10.48 = 3.22 lb. absolute pressure in the condenser.

446. Q.—If vacuum drops from 22 to 15 in., what will be the corresponding reduction in power?

A.—The amount that the reduction of your vacuum from 22 in. to 15 in. will reduce the power depends on the conditions under which the engine is running. Twenty-two inches is the equivalent of 7.97 lb. absolute pressure, and 15 in. is the equivalent of 14.95 lb. absolute; so that this reduction is practically equivalent to increasing the back pressure in the low-pressure cylinder 7 lb. This will reduce the mean effective pressure in that cylinder by the same amount. Of course, in actual practice, the high-pressure cutoff will be increased by the governor enough to take care of this loss in power, but that will be at the expense of an increased use of steam.

The simplest way to treat this problem is considering all power to be developed in the low-pressure cylinder. Then the mean effective pressure will be equal to a constant times the initial pressure, in pounds absolute, times (1 + hyperbolic logarithm of r); the whole divided by r, and from the quotient subtract the back pressure.

In this case, r is the total ratio of expansion in the two cylinders, and is found by dividing the initial pressure by the pressure at exhaust opening in the low-pressure cylinder. The constant is a value to take care of the loss in pressure, due to drop between the two cylinders and the rounding off of the card at the corners, and will vary from 0.75 for an engine having a big drop between cylinders and the corners a good deal rounded, to 0.9 for an engine having little drop and where the corners are nearly square.

447. Q.—If a hole appears in a barometric condenser, what is the result?

A.—A hole in the barometric condenser, if it comes in the condenser head, will, of course, tend to reduce the vacuum and decrease the suction. If it is large enough, it may destroy the vacuum entirely. Ordinarily, the water is drawn into a barometric condenser by the vacuum in the condenser. The vacuum is created by the downfall of the water in the tail pipe. If the hole is in the tail pipe, it will allow air to draw in and mix

with this column of falling water, and will, of course, decrease the vacuum in the condenser head.

448. Q.—If you were running a siphon condenser and you lost the vacuum, but were getting the injection of water all right, where would you look for the trouble?

A.—Probably the condenser has flooded and the vacuum breaker has opened; or else the discharge from the tail pipe has got stopped up.

449. Q.—The temperature of the cooling water is 55 deg. F. It is heated to a temperature of 120 deg. by passing through the condenser, the temperature of the condensed steam is 135 deg. The steam is exhausted into the condenser at 3 lb. absolute. Calculate the quantity of condensing water per pound of steam.

A.—Weight of water necessary for condensing purposes equals the heat lost by the steam divided by heat gained in condensing water, or, put in the form of an equation,

$$W = \frac{H - T_0 + 32}{T_2 - T_1}$$

W = pounds of cooling water necessary to condense and cool one pound of steam to the required discharge temperature.
H = total heat of steam at condenser pressure above 32 deg.
T_0 = temperature of condensed steam at the discharge.
T_1 = the initial temperature of the cooling water.
T_2 = the final temperature of the cooling water.

From steam tables, H for steam at 3 lb. absolute, as per your example, equals 1121.6. $T_0 = 135$ deg.; $T_1 = 55$ deg.; $T_2 = 120$ deg.

Then, $W = \frac{1121.6 - 135 + 32}{120 - 55} = 15.7$ pounds of condensing water per pound of steam.

Actually more than this will be needed, about 20 lb. per pound of steam being usually required.

450. Q.—In a condenser connected with a dry vacuum pump, how are the gases and water taken care of?

A.—A dry vacuum pump is used to remove air or gases from the top of a condenser so that it will not be necessary to remove them with the hot well pump. This enables the hot well pump to be much smaller, which is usually an advantage, as it is located below the condenser, and it also permits the maintenance of a greater vacuum than would be possible if all the air and gases were pumped out with the water. The hot well pump removes the water from the bottom of the condenser and lifts it to the feed water heater or feed water tank. The dry vacuum pump does not actually pump any of the condensate, its chief duty is to assist the hot well pump in maintaining vacuum.

451. Q.—Can copper tubes be used in connection with surface condensers? If so, why are they not more generally employed?

A.—Copper can be, and is, used for condenser tubes, although when salt is present in the circulating water, more or less trouble is experienced on account of the resulting corrosion. Under normal conditions it is undoubtedly due to the relatively high cost of copper that this material is not used exclusively for condenser tubes.

452. Q.—In surface condensers, are both ends of the tubes packed, or is one end fixed to the tube sheet?

A.—In single tube condensers, one end of the tube is screwed or expanded into the tube sheet, and the other end is packed. In some condensers the tubes are corrugated, which gives them additional area, and they are either expanded into the head at one end, and expanded into a bushing at the other, or expanded into both heads.

453. Q.—In a surface condenser, how would you know when the tubes were leaking?

A.—By putting on the full amount of circulating water and maintaining a vacuum in the condenser and measuring the amount of water collected from the wet vacuum pump when no steam is admitted to the engine. On surface condensers up to 1500 kw. capacity, a small leakage of this sort may exist. On larger units, where tubes are long and steam velocities high, the leakage has been found to vary with the load on the engine or turbine.

If any considerable amount of leakage is found, the condenser should be filled with water and the leaks located and stopped.

454. Q.—In the case of a surface condenser which can readily be shut down for the purpose, how could you detect leaks?

A.—If it is possible to shut down the condenser, a leakage test may readily be made by removing the two end heads of the shell and placing the inside of the condenser under a hydrostatic pressure. Any leakage which may exist will very readily show up at the ends of the tubes where they enter the end plates.

455. Q.—Where salt water is used for circulating purposes, how can the presence of this be detected in the condensate? In other words, how can leakage of the circulating water be discovered by testing the condensate?

A.—Ten cubic centimeters of the condenser water are measured out by a pipette, placed in a beaker, and two or three drops of potassium chromate solution added. A standard solution of silver nitrate is then added drop by drop from a burette, but only until the liquid in the beaker maintains a faint pink color on agitating. The volume of silver nitrate solution used is then recorded. Ten cubic centimeters of the sample of circulating sea water are now measured out, 90 cu. cm. distilled water added, mixed, and 10 cu. cm. of the mixture taken. A few drops of the chromate solution are now added, and the silver nitrate added as before, the amount of the latter used being recorded. The amount of silver nitrate used for the circulating water multiplied by ten and divided into the amount used for the condenser water and the quotient multiplied by 100, gives the percentage of sea-water leakage in the condensate. The silver nitrate solution usually used is made up by dissolving 9.6 gm. silver nitrate in 2 liters of water. This is of such strength that 1 cu. cm. is equivalent to one milligram of chlorine.

456. Q.—In starting an independent jet condenser, is engine or condenser started first? And which is shut down first?

A.—The practice in starting up a condensing engine

with independent jet condenser varies according to the nature of the load, and the conditions for starting. If an engine has to start under load, the condenser should be started first, so that the power due to the vacuum may be available in helping the engine pull up to speed. If an engine is started without the load on, it is usually better to start up the engine and let it warm up non-condensing, exhausting through the relief valve to atmosphere, then start up the condenser, and as the load is thrown on, put the vacuum on the engine. The engine should not be started up connected to the condenser, with the condenser idle, and vacuum should always be pulled in the condenser before the engine is allowed to exhaust into it, otherwise there is possibility of the water from the condenser flooding back into the exhaust pipe and even reaching the cylinder, which would, of course, be disastrous. Also pressure may be put on the condenser and burst it.

In shutting down, the engine should be shut down first, for this same reason, so that the engine will stop while there is still vacuum on the exhaust system, to keep water out of the exhaust pipe and cylinder. After the engine is stopped, the condenser can be stopped and the vacuum broken, if desired. In any case, the supply of injection water should be shut off before the vacuum pump is stopped, so that the condenser will be pumped clear of any surplus of water to avoid danger of getting it into the engine cylinder.

456. Q.—Is THERE danger of getting water into the cylinder while using a surface condenser?

A.—The possibility of getting water into the cylinder of an engine working in connection with a surface condenser is quite remote. The conditions under which this could happen would be in case the condensate pump should stop, allowing the condenser to flood. There should be some form of atmospheric relief valve cut in between the engine exhaust and the steam inlet of the condenser, so that should anything of this nature happen, the exhaust from the engine would pass out to the atmosphere.

458. Q.—Is THERE danger of getting water into the cylinder when using a siphon condenser?

A.—Siphon condensers, and all condensers of the jet type, are generally provided with some form of auto-

matic vacuum breaker, so that in case the condenser should become flooded, with the possibility of allowing some of the water to enter the cylinder of the engine or the turbine, the vacuum would be broken, and thus avoid this trouble. The possibility of getting water into the cylinder, when using a siphon condenser, is somewhat greater than when using a surface condenser, especially if cooling water is pumped to the condenser head.

Piping

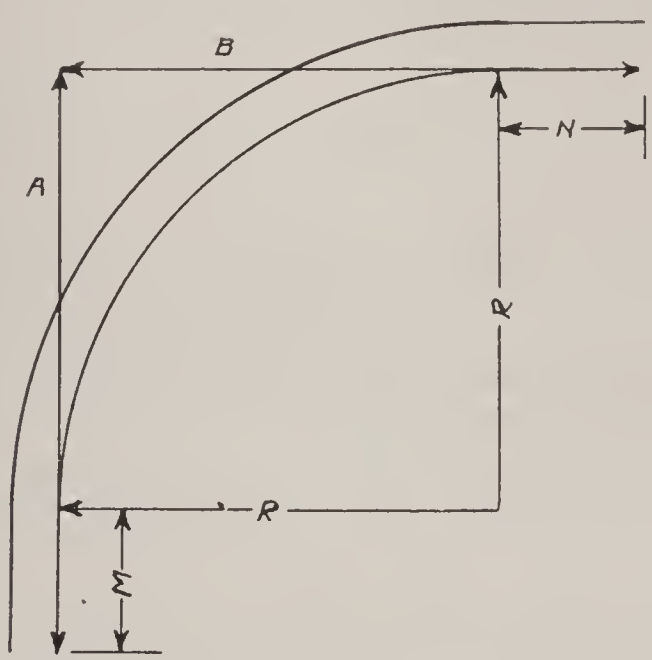
459. Q.—How is the size of steam pipe for an engine figured?

A.—The following formula gives good results in the calculation of steam piping for engines:

$$\text{Area of pipe, sq. in.} = \frac{\text{Area of cylinder sq. in.} \times \text{piston speed, ft. per min.}}{\text{Mean velocity of steam in pipe, ft. per min.}}$$

Common practice gives the exhaust pipe approximately twice the area of the steam pipe. Mean allowable velocity in the steam pipe is usually taken as 6000 ft. a minute.

460. Q.—How is the length of pipe needed for a 90-deg. bend found?



A460. Length for 90-deg. bend

A.—Use the following formula, the letters referring to distances indicated in Fig. A460:

$L = M + N + (3.1416\ R \div 2)$

L = length needed in inches

$M = A - R$; $N = B - R$

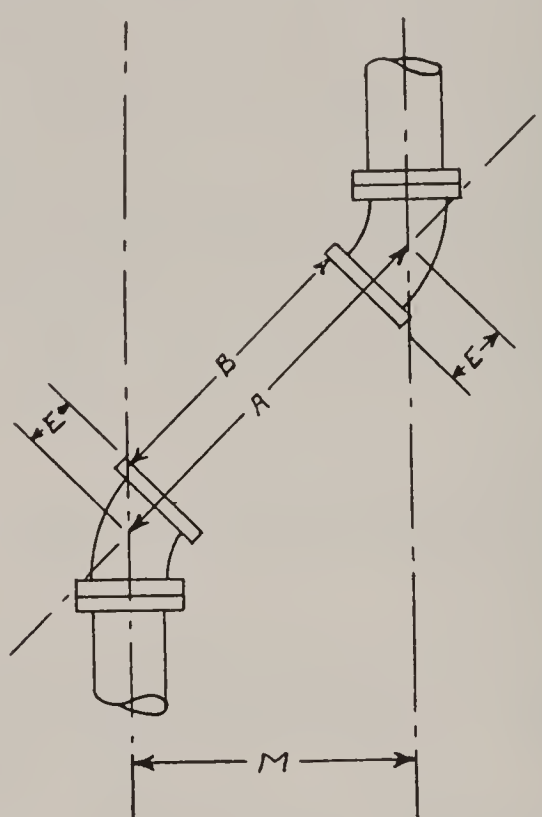
A and B are the square distances from one pipe to the other, measured along the *inside* of the pipe.

R is the radius of the bend.

Allowance must be made for make up into fittings.

461. Q.—What is the length of pipe needed for a 45-deg. offset?

A.—To find the net length of pipe for any 45-deg. offset, measure perpendicular distance M between lines, Fig. A461. Multiply this distance by 1.414. Subtract



A461. LENGTH OF 45 DEG. OFFSET

from this quotient twice the distance E which is the length from the center to face of a standard 45 deg. ell. The remainder is the distance between faces, or B.

For screw fittings, proceed as before, but add to the remainder, the length of thread entering the two fittings.

462. Q.—For a 36 by 48-in. Corliss engine at 60 r.p.m., what are the diameters of steam and exhaust pipes?

A.—Using the formula of A459, the area of a 36-in. cylinder is 1018 sq. in. With a 4-ft. stroke, at 60 r.p.m., the piston speed will be $60 \times 2 \times 4$ or 480 ft. per minute. Allowing a velocity in the steam pipe of 6000

ft. a minute, the area for the steam pipe would be 1018 $\times$ 480 $\div$ 6000 or 81.4 sq. in., which would be a little over a 10-in. pipe. With an exhaust velocity of 4000 ft. a minute, the area for exhaust pipe would be 1018 $\times$ 480 $\div$ 4000, which would give 122 sq. in., which would be a little under a 13-in. pipe.

463. Q.—What size steam pipe would be needed to carry 1.9 lb. of steam per minute at 70 lb. pressure, velocity of 5000 ft. per min.?

A.—The simplest and most common formula for the flow of steam through pipes is based on an allowable steam velocity of 6000 feet per minute, friction and other causes of drop in pressure being disregarded. Thus, for a velocity of 6000 ft. per minute, we have,

$$d = 0.175\ (W \div y)^{\frac{1}{2}}$$

Where d = the diameter of the pipe in inches;
y = the density of the steam in pounds per cubic foot;
W = the weight of steam flowing, in pounds.

For the data given,

$d = 0.175\ (1.9 \div 0.19)^{\frac{1}{2}}$
$= 0.175\ (10)^{\frac{1}{2}} = 0.55$ in.

For velocity of 5000 ft. a min. the diameter would be increased in proportion to the square root of the velocity, or, would be,

$$0.55 \times (6 \div 5)^{\frac{1}{2}} = 0.6 \text{ in.}$$

This is inside diameter, and would correspond to ½-in. pipe.

The value of y is found for 70 lb. pressure from a table of "Properties of Steam".

464. Q.—What causes water hammer? Is it dangerous?

A.—Water hammer is usually caused by the rushing of a slug of water through a pipe, fetching up against an obstruction such as a sharp right angle turn, a valve, or a return bend. The rushing through is caused by the condensation of steam ahead of the water, causing a vacuum into which the slug of water rushes. Water hammer, if it is extreme, is likely to be dangerous, and often breaks cast-iron fittings and valves.

465. Q.—Should a steam line be inclined toward the engine or toward the boiler?

A.—Incline the steam line toward the engine, and put a good separator at the engine to take out the moisture. This moisture will not drain back to the boiler against the flow of the steam.

466. Q.—What is necessary for an efficient steam separator? Where should it be placed?

A.—A steam separator is a device to be incorporated in the steam line, whether live or exhaust steam, for the purpose of removing in transit the particles of condensation which the steam contains.

The condensation is thrown out by centrifugal force, changing direction of the flow, by scrubbing, passing over a series of baffles, or by slowing down the flow, letting the moisture precipitate. A combination of two of these methods is often used.

On steam lines leading to engines, etc., the separator is placed as close as possible to the throttle, usually right above it, and the condensation removed by means of a steam trap located below the level of the separator. On exhaust steam lines where the steam is used for heating or other purposes, an oil separator is used, being located where it can best do its work, a grease trap taking care of the oil and water. If there is no back pressure, the trap can be dispensed with and a drain in which there is a loop that will prevent the escape of anything but the oil and water run direct to the sewer.

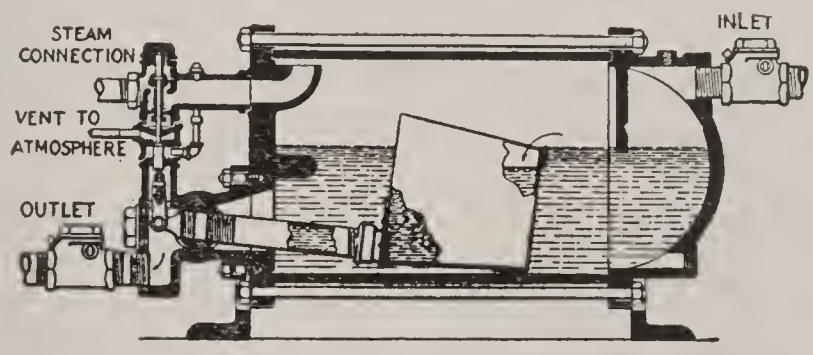
467. Q.—Why is an oil extractor used in an exhaust line, when the condensate is fed to a boiler? Why for supplying a heating system?

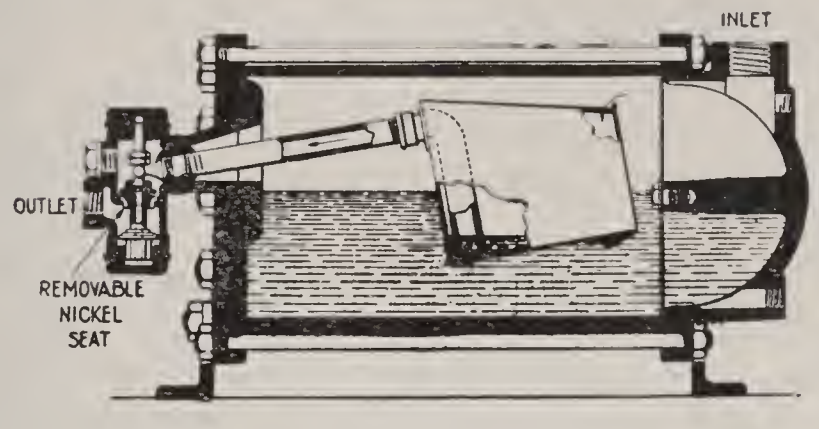
A.—Oil in a boiler is liable to coat the plates above the fire and cause overheating. It is, therefore, dangerous. In a heating system, the oil coats the inner surfaces of pipes and radiators, and makes them less effective as transmitters of heat. Oil extractors are used to remedy these evils.

468. Q.—What is a steam trap?

A.—Traps are devices used on steam lines or systems for the purpose of removing condensation which may collect and do damage to the piping or apparatus used in connection therewith. This is done, or should be done, when the trap is in working condition without the loss of steam.

Steam traps may be divided into two classes, depending on their use,—return and nonreturn. The former receive the condensed steam and return it to a boiler having considerably higher pressure than that acting on the returns. Nonreturn traps are used where the water of condensation is not returned to the boiler but is discharged into a sewer or a hot well.





A468. RETURN AND NONRETURN TRAPS

469. Q.—What are some of the causes of a return trap not working?

A.—Valves becoming clogged with sediment, or the trap becoming air bound are the principal causes of faulty operation of steam traps.

470. Q.—On any trap, when changing from low pressure to high pressure, why is it necessary to put in a smaller seat and a smaller valve?

A.—The force to close a trap is either the expansion of a tube or rod, or the weight of a float or bucket. The force to be overcome is the area of the valve seat times the steam pressure, and with low pressure, this total force to be overcome will be a good deal less, for the

same area of valve seat, than with high pressure, hence with high pressure in the trap, the area of the valve seat must be made less, or the mechanism operating the valve will not be able to open the valve.

471. Q.—How does a Bundy trap work?

A.—The Bundy steam trap is used to return the water of condensation directly back to the boiler from the receiver or reservoir. The trap must be placed at such a height that the water will flow into it by gravity. This distance is usually 3 or 4 ft. above the boiler water line.

When the trap is full the receiving bowl tilts and pulls open the steam valve for the admission of live steam under boiler pressure. The pressure on trap and boiler are thus equalized, but the trap being 3 ft. above the water line, the water will pass by force of gravity from the trap back into the boiler. When empty, the receiving bowl tilts back to the filling position and is ready for the next charge of water.

472. Q.—If a steam trap should get out of order, how can it be repaired without shutting down the system?

A.—It should not be necessary to shut down the system in order to repair a steam trap.

In the line joining the system and trap should be placed some form of shutoff valve allowing the trap to be isolated from the system, and a bypass leading around the trap.

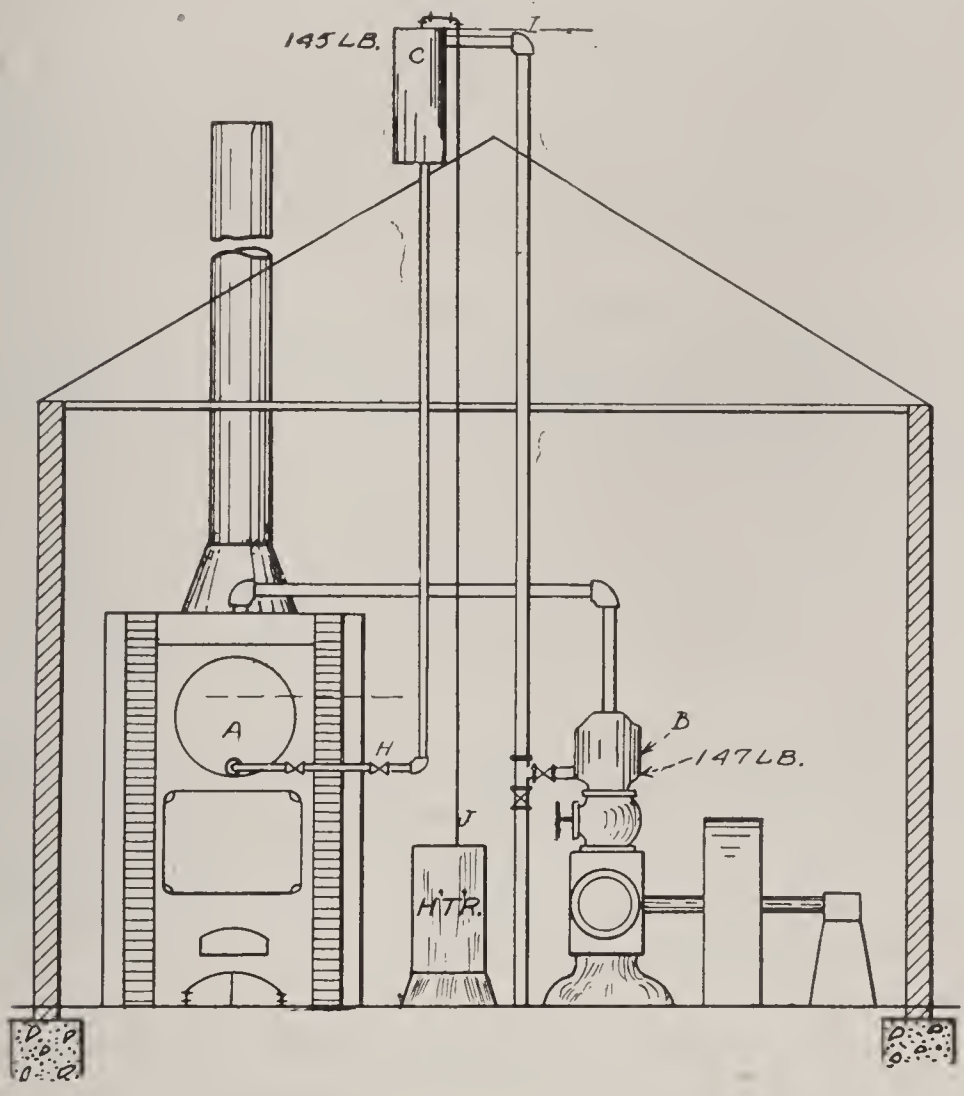
When a trap is in need of repairs, this valve may be closed and the trap taken apart; but to insure reasonable drainage of the system, the condensation which may collect should be removed at frequent intervals by opening the bypass until dry steam appears. This will do until the trap can again be placed in operation.

473. Q.—What is the principle of a Holly loop?

A.—If a closed vessel be placed above a boiler and connected to it by two pipes, one to the steam space and the other below the water line, the pressure in the closed vessel will be the same as in the boiler, and the water in the pipe connected below the water line will stand at the same level as the water in the boiler. If a vent pipe on the closed vessel be opened the pressure in the closed ves-

sel will be decreased and the water will rise in the one pipe high enough to balance the difference in pressure between that in the closed vessel and that in the boiler, and steam will flow through the other pipe into the closed vessel in its effort to equalize the pressure between the boiler and the closed vessel.

When the closed vessel is a discharge tank, and steam rising to the tank holds water in suspension, this water



A473. HOLLY STEAM LOOP

will fall to the bottom of the tank and into the return pipe which is connected below the water line of the boilers and increase the head or height of water in this line. This extra head of water overbalances the increase in height due to the difference in pressure between the boiler and the discharge chamber, and water will flow into the boiler until the proper level is reached. This action causes the slugs of water and steam, or rather a spray of water and steam, to rise from the receiver to the discharge chamber, where separation occurs, the water falling, to the return pipe, and the steam

condensing with a little of it blown out of the vent pipe to reduce the pressure. To further aid the operation, a check valve is placed near the boiler in the return line, which will prevent the water of the boiler from rising in the return line should the other side of the system break or close.

474. Q.—What is the wall thickness of 6-in. standard wrought iron pipe and that of 6-in. double extra heavy wrought iron pipe?

How do you calculate the bursting pressure of pipe?

A.—The wall thickness of standard 6-in. wrought iron pipe is 0.28 in. while double extra heavy wrought iron pipe of like diameter has a wall thickness of 0.864 in.

The bursting pressure of pipe of given wall thickness is equal to the quotient obtained by dividing the product of twice the thickness in inches, and the tensile strength of the material by the internal diameter of the pipe in inches.

To determine the working pressure, divide by a suitable factor of safety which may be assumed as 6 or 8.

475. Q.—Does steel or iron resist corrosion better, and why?

A.—Wrought iron usually resists corrosion better than steel, because in iron there is no carbon, hence the chemical action does not take place so rapidly. There is, however, considerable dispute upon this point, and the results of experience do not agree.

Reciprocating Pumps

476. Q.—How many valves are there in the steam end of a duplex pump?

A.—The common duplex pump as usually found in the every day plant has but one valve for each steam cylinder. In the smaller sizes both valves are located in one steam chest common to both cylinders.

477. Q.—What is the action of springs in the water end of a pump?

A.—The springs in the water end of a pump are used simply to close the valves quickly, and prevent slippage, or leaking of water through the valves. If there were no springs, at the reversal of stroke when one set of

valves must close and the other open, the closing valves might act sluggishly, and thus permit some water to slip back through them before they seated. The springs are for the purpose of making this action more rapid, and insuring a quick, firm closing of the valves.

478. Q.—Why are the valves of a duplex pump given lost motion?

A. Lost motion on the valve of a duplex pump allows the valve to remain motionless most of the stroke, thereby permitting the piston to make the full stroke before the valve is reversed.

Too little lost motion on steam pump valves will shorten the stroke, and vice versa.

479. Q.—If one of the spools on a duplex pump should slip, how would the pump act? How can the proper location of the spool be determined?

A.—In case of a spool on one of the rods of a duplex pump slipping, the steam supply on the opposite sides will be unbalanced, with the consequent uneven operation of the pump. On one stroke steam would be admitted longer than on the other, resulting in a non-uniform movement.

The rod is usually marked in some way, such as a chisel mark, or if the pump has been in service for some time, an accumulation of dirt on the rod next to where the spool has been will indicate its proper position.

However, the better and most positive way to reset the spool is to place the steam and water pistons at the center of travel, and then shift the spool until the rocker arm is in exact vertical position, as shown by a plumb line. Then tighten the spool in place.

480. Q.—How much lap has the valve of a duplex pump and why is it used?

A.—The steam valves of a duplex pump have just enough lap to make them steam tight, as they are designed to work the steam full stroke.

481. Q.—How are duplex pump valves set?

A.—Setting the valves in a duplex pump consists in adjusting the lost motion so that when the piston on one side has just completed its stroke, it has moved the steam valve of the other piston to its extreme position; then push the piston to its other extreme position, and

adjust the lost motion so that the valve is in its other extreme position. Repeat the operation for the other side of the pump. These pumps work the steam the full length of the stroke, so that about all that can be done is to see that the pistons work full stroke.

482. Q.—What defect of a duplex pump may cause steam to be cut off before the end of the stroke? How can it be remedied?

A.—A duplex pump will short stroke, as it is called, if there is not enough lost motion in the valves. The remedy is to adjust the lost motion until proper action is secured.

483. Q.—Do you ever have to take up the wear in the valve gear of a duplex pump?

A.—As wear occurs, it is necessary to take up on the valve gear of a duplex pump, but at proper speeds of running the wear is slow.

484. Q.—Why is an air chamber used on a pump handling cold water?

A.—The function of an air chamber on a pump is to take care of shocks due to the surging of the water in the discharge or suction lines of the pump. This chamber is filled with air, so that as the column of water surges back and forth instead of straining the various parts of the pump and the suction and discharge lines, it will act on this contained air, compressing it, and allowing it to expand with the release of the force of this surge.

485. Q.—Why is the water cylinder of a pump longer than the steam cylinder?

A.—A water cylinder on a steam pump is usually longer than the steam cylinder for the reason that the water piston is usually longer than the steam piston, because a water piston is packed with soft packing, and the wider surface of this packing is needed to prevent leakage by the piston and to reduce the wear thereof.

486. Q.—What would you do with a duplex pump, one of the pistons of which was continually striking the cylinder head?

A.—A piston striking the head of a pump means that

the stroke is too long. The lost motion in the valve action should be decreased.

487. Q. What is the remedy for short stroke in a duplex pump?

A.—The most common cause of short stroking is the packing too tight. First ease up on the piston rod packing, and if that does not remedy the trouble, examine the valve lost motion to see whether this is set correctly.

488. Q.—With one side of a duplex pump out of commission, how would you keep the pump in operation?

A.—Whether a duplex pump could be kept running with one water cylinder disabled, would depend on just where the difficulty lay. If the trouble was in the piston, it would be possible to take the piston off the rod and simply let the steam piston work back and forth, setting up the stops controlling the steam valve on that side, so that the pump would make a very short stroke; just enough to operate the valve on the other side of the pump.

489. Q.—How would you determine whether the plunger or piston of a steam pump leaked without opening the cylinders?

A.—If there is a bad leak of water past the piston or plunger of a pump, it would be necessary to drive the pump at a good deal higher speed than normal in order to get the required service. If there is a water meter in the line, the reading of this meter can be compared with the computed water pumped, but a difference here may be due to leakage past the piston or through the valves. A bad leak can usually be heard by listening close to the water cylinder. If the leak is small, or if the pump is running very slowly, this cannot probably be detected.

490. Q.—What are the relative advantages of simplex and duplex pumps?

A.—It does not seem that the relative advantages of simple and duplex steam pumps have been very definitely determined. The simplex pump, since it has a single pump cylinder only, is liable to give more or less fluctuation in pressure in the discharge line, particularly

if the pump is running slow. It is, however, more economical of steam than the duplex, and somewhat cheaper in first cost. The duplex pump has the advantage in steady pressure supplied, also can be run at a slower speed than the simplex pump.

491. Q.—How MANY boiler horsepower will be required to drive a 10 by 8½ by 12-in. duplex pump working against a total head of 100 ft.?

A.—The theoretical horsepower required to raise the water with a 10 by 8½ by 12-in. duplex pump against a pressure head of 100 ft., would depend on the amount of water pumped per minute. Assuming a piston speed of 100 ft. a minute or 100 strokes a minute, the pump would deliver

$$\frac{0.7854 \times 8.5^2 \times 12 \times 100}{1728} = 39.4 \text{ cu. ft.}$$

or $39.4 \times 62.5 = 2462$ lb. per minute.

$$\text{Then hp.} = \frac{\text{weight of water per minute} \times \text{height}}{33{,}000}$$

$$= \frac{2462 \times 100}{33{,}000} = 7.47 \text{ hp.}$$

Due to leakage past the valves, a certain amount must be allowed for "slippage." Also, a certain amount is allowed for the loss in the steam end of the pump. Thus, with an assumed combined pump efficiency of 60 per cent, the actual horsepower required would be

$$\frac{100 \times 7.47}{60} = 12.45 \text{ hp.}$$

This value is, of course, in terms of steam cylinder horsepower. There is no definite relation between this horsepower and boiler horsepower. Ordinarily a steam pump may use about 120 lb. of steam per horsepower-hour. Thus, $120 \times 12.45 = 1495$ lb. of steam per hour. Dividing this by 30, the pounds of water evaporated per boiler horsepower per hour, gives 49.8 boiler horsepower.

492. Q.—How HIGH will a pump with an 8-in. steam piston and a 6-in. plunger force water with 80 lb. of steam?

A.—The rule is, that the water pressure times the area of the water piston is equal to the steam pressure times the area of the steam piston.

Then $X \times (6^2 \times 0.7854) = 80 \times (8^2 \times 0.7854)$;

$X \times 36 = 80 \times 64$

$X = 80 \times 64 \div 36$

$X = 142.2 =$ water pressure lb. per sq. in.

$0.433 =$ pressure exerted by 1 ft. head.

Then $142.2 \div 0.433 = 328 =$ head in feet.

This is neglecting friction, so allowing 10 per cent for friction, we would have a head of $328 - 32.8 = 296$ ft., nearly, the height to which the pump would force water.

493. Q.—How MANY gallons of water would be delivered by an 8 by 10 by 12-in. duplex pump at 100 strokes per minute; piston rod being 3 in. diameter?

A.—For a duplex pump, the plunger displacement is equal to

$$10^2 \times 0.7854 \times 12 = 942.48 \text{ cu. in.}$$

This is the plunger displacement of the head end. That on the crank end is equal to

$$942.48 - (3^2 \times 0.7854 \times 12) = 857.65 \text{ cu. in.}$$

Therefore, for each cylinder, there is delivered per minute, when operating at a speed of 100 strokes per minute. $(50 \times 857.65) + (50 \times 942.48) = 90{,}007$ cu. in. per cylinder; or $2 \times 90{,}007 = 180{,}014$ cu. in. of water delivered by the pump per minute. As there are 231 cu. in. in a gallon, the number of gallons delivered will be equal to $180{,}014 \div 231$, or approximately 780 gal. per minute. In these figures slip and efficiency have not been taken into consideration.

494. Q.—WHAT CAUSES quick motion in a vacuum pump at the beginning of the stroke?

A.—Quick motion at the beginning of the stroke of a vacuum pump which becomes slow towards the middle of the stroke is on account of a lower vacuum at the beginning of the stroke. Unlike a pump which pumps a liquid, the vacuum pump handles air, or rather exhausts it from some other appliance, such as a heating system or surface condenser.

When the piston approaches the end of a stroke, it forces out through the valves a certain amount of air under pressure, while at the beginning of the next stroke

there is nothing to retard the rapid action of the piston until a vacuum is established in one end of the cylinder and pressure in the other, and as soon as this is done the piston is retarded.

495. Q.—Why can not a vacuum pump be used for boiler feeding purposes?

A.—The vacuum pump is not properly proportioned for use as boiler feed—that is, the steam cylinder is not large enough in proportion to the pump cylinder. Another thing, the valves on the pump end are usually not of the right material to stand high-pressure work, and the springs on the valves are not sufficiently strong to seat the valves quickly. The vacuum pump is usually designed without any air chamber on the pump end, and this is always a desirable thing to have in a pump which is to work under high pressure.

496. Q.—What is the difference between a dry and a wet-vacuum pump?

A.—The dry vacuum pump is used for removing air and gases from the top of a condenser, so that they will not have to be taken out by the hot well pump at the bottom. This enables the hot well pump to be much smaller, which is usually an advantage, as it has to be located below the condenser, and also it permits of maintaining a greater vacuum than would be possible if all the air and gases were pumped out with the water. The air and permanent gases will tend to rise to the top of the condenser, and it is difficult to draw them down with the condensed steam to the bottom.

The wet vacuum pump, or hot well pump, is used for taking away the condensed steam from the surface condenser, or in the case of a jet condenser, it takes away both condensed steam and cooling water. The dry vacuum system is, however, more often used with the surface type.

With the jet condenser, usually only one pump is used, and that is called simply the "vacuum pump," and handles both air and water. The cooling water being sprayed into the steam space of the condenser in the jet type, picks up the air and gases and carries them away in solution.

497. Q.—We are operating a 10-ton ice plant. The water for cooling the ammonia and for condensing steam

is supplied from a well by a 6 by 5¾ by 6-in. duplex pump. The pump is down in the well, 20 ft. below the surface and within about 15 ft. of the water. From the pump the water is raised 60 ft. Normally the pump makes about 40 strokes a minute and is run only when the ice machinery is in operation. What is the approximate cost per 1000 gal. of running this pump with steam at 100-lb. pressure, soft coal at $4.50 per ton being used for fuel? The steam is supplied by the ice plant boiler.

Our well has given out and we must either put down a new well or buy our water from the local water plant. The question is, which would be the more economical? The cost of a new well would be about $300.

Under the above circumstances, what could we afford to pay the city per 1000 gal. of water?

A.—Some computations on probabilities in your plant will give an indication of what you are after.

At 40 strokes per min. in each cylinder, your duplex pump would be handling about 50 gal. per min., and allowing 80 ft. gross head, which is 5 ft. for pipe and valve friction, this pump will require about 0.9 hp. on the steam end.

Allowing 120 lb. of steam per hp.-hr., this will call for 108 lb. of steam per hr., or about 0.036 lb. of steam per gal. of water pumped. For 1000 gal. this would give 36 lb. of steam.

Assuming that 7 lb. of steam are evaporated per lb. of coal burned, this would give about 5 lb. of coal used per 1000 gal., which, with coal at $4.50 a ton, would make a cost of 1.128 c. per 1000 gal. for fuel.

With a well costing $300, the interest, upkeep and depreciation would amount to about $50 a yr., and probably about the same amount would be necessary for the interest, depreciation and upkeep allowance on the pump and piping. If 2,500,000 gal. are pumped per yr., this would be a cost of about 4 c. per gal. for the well, pump and piping, or a total of about 5⅛ c. per 1000 gal. for fuel and upkeep, no extra labor being required in the plant, on account of the use of the well and pump.

With these assumptions, which are probably not far from correct, you could pay any price up to 5 c. per 1000 gal. for the city water and it would be profitable to do so. Beyond this amount, it would probably be cheaper for you to install a new well and pump your own water.

498. Q.—What kind of a pump would be used to get expansion of the steam?

A.—In order to get expansion of the steam in a pump, one of two kinds must be used, either a flywheel pump, which will have, by means of the flywheel, some means of operating the valves before the end of the stroke, and permitting the storing and restoring of energy during the remainder of the stroke, or a compound pump, where the steam may expand from one set of cylinders into a succeeding set.

499. Q.—What is the difference between a high and a low-duty pump?

A.—Duty is a term defining the work done by a pump for a given weight of steam. If the steam is used economically, as in a compound condensing engine, the pump is called high-duty, while the common direct acting pump is of the low-duty type. The term duty is often erroneously applied instead of the term service to denote the character of the work done. Light service pumps have comparatively large water cylinders and are used for pumping against pressures lower than that in the boiler furnishing the steam, as in tank and house service. In heavy service pumps the water cylinder is smaller than the steam cylinder and the pump is used for boiler feeding and other high-pressure work.

500. Q.—What is meant by the percentage of slip of a pump?

A.—In any pump the amount of liquid thrown each stroke does not equal the full volume of the piston displacement, because the valves do not open and close instantly. When the piston starts on its suction stroke, the discharge valves do not seat at once, and a little water pulls back into the cylinder from the discharge chamber. When the piston starts on the pressure stroke, the inlet valves do not close immediately, and a little water slips back from the cylinder into the suction pipe. The loss from these two causes is called the percentage of "slip," and may vary from 5 to 25 per cent, depending upon the tightness of the valves and the speed at which the pump is running.

Centrifugal Pumps

501. Q.—How do you figure the horsepower and discharge of a centrifugal pump?

A.—It is assumed that the question refers to a pump already built and in service. The horsepower can then be ascertained by measuring the number of pounds of water per minute being pumped and the pressure head at the discharge nozzle in feet. These multiplied together and divided by 33,000 give the net horsepower. The gross horsepower to drive it will be greater, as some allowance must be made for various hydraulic and mechanical losses, which vary in amount depending on the size and make of the pump, and are generally expressed in terms of efficiency. Thus, a pump having a guaranteed efficiency of 70 per cent would have losses amounting to 30 per cent; and after ascertaining the net horsepower, you would have to multiply by $100 \div 70$ to get the gross horsepower required. This gross horsepower will; of course, be the horsepower that would have to be supplied to operate the pump.

502. Q.—What horsepower motor is required to drive a centrifugal pump at 500 r.p.m. against a head of 40 ft. and suction lift of 5 ft.; suction and discharge being 3 in. diameter, with six elbows in the discharge line?

A.—Before the horsepower of the pump motor can be determined, the pump capacity in gallons or cubic feet per minute must be known. The diameter of the discharge end determines the amount of water handled, and for a 3-in. discharge would be about 200 g.p.m. The rule is that

G.p.m. = width of impeller, in. $\times$ circumference of impeller in ft. squared $\times$ r.p.m. $\div$ (square root of head in ft. $\times$ 110,800).

Speed will depend on the head against which the pump operates, r.p.m. varying inversely as the square root of the head. The head must be the sum of the suction and forcing heads. The value 110,800 is a constant derived from tests, and the discharge opening is designed to take care of the capacity thus found at a velocity of discharge of 9 to 15 ft. per second.

Power required to pump the water is:

$$\text{Hp.} = \text{g.p.m.} \times 8.33 \times \text{h} \div 33{,}000$$
$$= \text{g.p.m.} \times \text{h} \div 3960$$

Where g.p.m. is gallons per minute, and h is total head in ft.

Average pump efficiency E is about 70 per cent, and power required to drive the pump would be.

$$\text{Hp.} = \text{g.p.m.} \times \text{h} \div (3960 \times \text{E})$$

The six elbows will have a resistance at water velocity of 10 ft. per second of $\text{h} = 0.98\ \text{v}^2 \div 64.32$ per elbow, or for six elbows, 9.15 ft., so that the total head will be $40 + 5 + 9.15 = 54.15$ ft.

Motor horsepower will then be,

$$\text{Hp.} = 200 \times 54.15 \div (3960 \times 0.70)$$
$$= 3.91 \text{ hp.}$$

Allowing for slip and motor losses about a 4.5-hp. motor should be used.

503. Q.—We have a 50-hp. Standard Westinghouse Engine, supplied with steam from a 100-hp. boiler fed by means of a Metropolitan Injector. We intend to heat our feed water with exhaust steam in a National Feed-Water Heater and wish to inquire whether it would be more economical to employ a belt driven pump belted to the engine shaft or a direct-acting steam pump.

A.—As a rule, the belt-driven pump will be more economical than the direct-acting steam pump, so far as power is concerned. The steam pump uses a large amount of steam per hp.-hr., has the advantage that it can be regulated readily to meet the need for more or less feed water in the boiler.

The belt-driven pump must run practically at a constant speed, and the only way of regulating the water fed to the boiler is by a bypass valve which will allow more or less of the water pumped to flow back into the suction.

Another method would be to have some kind of a clutch between the engine shaft and the pump, so that it can be stopped and started as may be necessary to keep the water level correct in the boiler. As you probably have all the exhaust steam you need from the engine to heat the feed water, there is no advantage in having more exhaust steam from the pump.

504. Q.—What regulates the height to which a centrifugal pump can deliver water?

A.—This is controlled almost entirely by the speed and diameter of the impeller. Thus, we can pump water to the same height with a small impeller at high speed

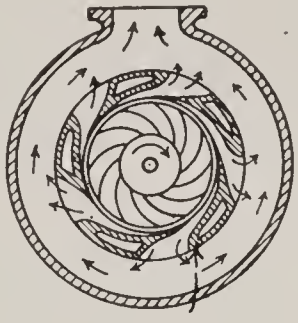
as we can with a large impeller at low speed, the linear velocity at the periphery of the impellers being the same.

505. Q.—In what way does the shape of the runner affect the pressure?

A.—The shape of the runner does not affect the pressure; that is, any shape runner can be made to give the desired pressure, but different shapes of runners will give different results if attempt is made to vary the pressure. Thus, an impeller with backward curved vanes will give an increased capacity at constant speed if the pressure is lowered; an impeller with straight vanes will give a variable capacity at constant speed with a practically constant pressure, and an impeller with forward curved vanes will give a decreased capacity at constant speed as the pressure is reduced.

506. Q.—What purpose do the diffusion rings in a centrifugal pump serve?

A.—The impeller imparts a high velocity to the water; but it is pressure that is desired rather than velocity, so some means must be provided for convert-



A506. SECTION OF PUMP WITH DIFFUSION RING

ing this velocity into pressure, and the diffusion vane is a most efficient way of accomplishing this. The vane, Fig. A506, is a gradually expanding opening receiving the water from the impeller at the smaller end and, as the water travels through the increasing area, its velocity is reduced and the energy contained must appear in some other form, and does so as pressure.

507. Q.—Please give the relative advantages and disadvantages of the turbine type of feed-water pump as compared with the reciprocating plunger type.

A.—The turbine feed-water pump has the advantage of steadier flow of water, generally better economy —that is, lower power consumption—than the reciprocating pump for a given amount of water pumped, and can be used more readily for drive from the steam turbine or the electric motor.

On the other hand, the reciprocating pump is more flexible—that is, the speed can be more readily regulated, and the amount of flow from the pump to the boiler more readily controlled. It is a slow-speed rather than a high-speed machine, hence requires less attention, and perhaps has slightly less depreciation, although this is a point which might be questioned.

508. Q.—WHAT ARE the relative advantages of direct-connected motor drive and a steam turbine with reduction gears?

A.—This depends almost entirely on the available power and use for exhaust steam. Direct motor drive has better mechanical efficiency than geared drive, but the pump speed must be fitted to that of the motor. With geared drive, the pump may be chosen for best speed to suit the work, and turbine may be run at best speed for its steam pressure. If exhaust from the turbine can be used for heating, it will often show a better result—work per 1000 B.t.u.—than the direct motor drive which involves losses in generator, transmission and motor. Also, the turbine drive will often be cheaper to install.

509. Q.—WHAT WILL be the result if a centrifugal pump is started and the discharge valve then closed?

A.—If the discharge valve was closed after a centrifugal pump was started, the speed would probably be checked momentarily, and would then increase. This is due to the fact that no work would be done in lifting the water, the only work being that done against the friction of resistance of the water and that of the pump.

510. Q.—HOW WOULD you test the flow of water in the discharge from a 36-in. centrifugal pump through a 48-in. horizontal riveted steel pipe delivering 35,000 g.p.m. at a velocity of 373 ft. per min.? How high will it raise the mercury in a Pitot tube with a ⅜-in. round opening, facing against the current, the other opening

facing down stream? Should readings be taken from more than one point in the pipe to give correct results?

A.—For a 36-in. centrifugal pump delivering 35,000 gal. per min. through a 48-in. pipe.

Area of 48-in. pipe = 12.56 sq. ft.

For 35,000 gal. per min. at 7.48 gal. per cu. ft., the discharge = $35{,}000 \div (60 \times 7.48) = 77.9$ cu. ft. per sec. $77.9 \div 12.56 = 6.2$ ft. per sec. (mean velocity).

$$V = C\sqrt{64.4\ (h \div 12)\ (\text{sp.gr.} - 1)}$$

where V = velocity in ft. per sec.

C = coefficient of tube

$h \div 12$ = deflection of liquid in U tube measured in ft.

Sp.gr. = specific gravity of liquid in tube, which for mercury = 13.56.

Then $V = C\ 8.21\sqrt{h''}$, h'' being the deflection in inches.

For one type of double orifice Pitot tube, C = 0.84.

$$6.2 = 0.84 \times 8.21\sqrt{h''}$$

$h = 0.807$ or 13/16″ mercury (approx.)

The velocity should be determined at several points along a diameter of the pipe, as the velocity is greater at the center than near the sides.

Velocity per minute = $6.2 \times 60 = 372$ ft., which agrees closely with your figure.

511. Q.—Should water from the city mains be allowed to flow into the heater, thence pumped to the boilers, or should it be pumped through the heater?

A.—As to forcing water through the heater by city pressure, or taking the city water into the pump suction and pumping it through the heater, we would favor the latter plan. There is no advantage so far as power is concerned, one way or the other, but your pump will operate better when handling the water direct from the city mains than if handling the hot water from the heater. You can have a bypass around the heater from the pump, so that in case it is necessary to make repairs on the heater, the boiler can be fed direct from the pump and avoid shutting down the plant.

512. Q.—How does the efficiency of a centrifugal pump compare with that of a reciprocating pump?

A.—The mechanical efficiency of a centrifugal pump depends on the speed, design of runner, and head pressure and the restriction to the flow in the discharge pipe.

The efficiency ranges from 45 per cent to 80 per cent, depending upon the circumstances. The efficiency in a reciprocating pump depends principally upon the friction of the parts, the leakage of water through the piston packing, and slippage through the valves, which in turn depends on the pressure against which the pump is working, and the condition of the several parts. The efficiency may be as low as 60 per cent with the pump in poor condition, and as high as 95 per cent with all the parts in good shape. It is impossible to say what the efficiency of a given pump will be without knowing all the circumstances, but as an average may be taken as 65 per cent for the centrifugal, and 75 per cent for the reciprocating pump.

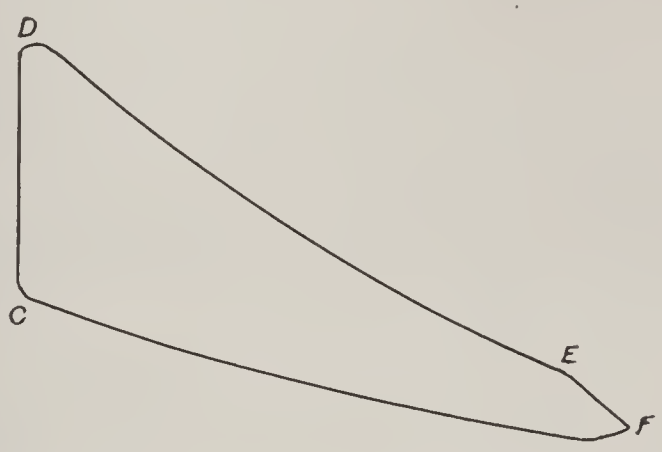
PART IV

Combustion Engines

Principles and Construction

513. Q.—What are the principles of operation of 4-stroke and 2-stroke cycle engines? What are the advantages of each?

A.—In the ordinary type of 4-stroke cycle engine, one ignition of gas takes place in one end of the cylinder every two revolutions of the flywheel, or every two double strokes. The following sequence of operations takes place during four consecutive strokes: (a) Inspiration of gas and air mixture during an entire stroke; (b) compression during the second (return) stroke; (c) ignition at or near the dead point, and expansion during the third stroke; (d) expulsion of burned gases during fourth (return) stroke.



A513. Diagram from 2-stroke cycle engine

In a 2-stroke cycle, single acting engine, an explosion takes place with every revolution or with each forward stroke of the piston. Referring to the diagram, Fig. A513, and beginning at E, when the exhaust port begins to open to allow the burned gases to escape, the pressure drops rapidly to F. Before the end of the stroke is reached, an inlet port opens admitting a mixture of gas and air from a reservoir in which it has been compressed. This mixture being under pressure assists in driving the burned gases out through the exhaust port.

The inlet port and the exhaust port close early in the return stroke, and during the remainder of the

stroke B C the mixture, which may include some of the burned gas, is compressed and ignition takes place at C as in the 4-stroke cycle engine.

In one form of the 2-stroke cycle engine, only compressed air is admitted while the exhaust port is open, the fuel gas being admitted under pressure after the exhaust port is closed. By this means, a greater proportion of the burned gases is swept out of the cylinder. This operation is known as scavenging.

The advantages of the 2-stroke cycle engine over the 4-stroke cycle engine are, that it requires a less number of valves, and that it makes twice as many working strokes in a given number of revolutions. At low speed, a 2-stroke cycle engine gives nearly twice the power of a 4-stroke cycle engine of like speed and size. At high speeds, however, a part of the fresh charge is apt to escape from the cylinder, and a portion of the spent charge is apt to remain. This results in a serious reduction in the power and efficiency of the 2-stroke cycle engine. The theoretical efficiency of the 2-stroke cycle is, however, exactly the same as that of the 4-stroke cycle having the same compression pressure, and the pressures, temperatures, volumes and work done are computed in exactly the same manner.

In the 4-stroke cycle engine, it is unnecessary to employ a closed crank case or a separate pump for compressing the gas and air.

514. Q.—What is the A. L. A. M. formula for horsepower? What does it give for an engine with four cylinders 4½ by 6½ in.?

A.—The A. L. A. M. formula for approximate rating of gasoline engines used in automobiles is the standard formula for the American Licensed Automobile Manufacturers Association. This formula is as follows:

Brake horsepower = diameter squared times number of cylinders, divided by 2.5. This formula is based on an assumed piston speed of 1000 ft. per minute, and takes no account of the stroke.

Using that formula to get the horsepower of your engine, we get,

$$\text{B.hp.} = 4.5^2 \times 4 \div 2.5 = 32.4.$$

The rating of your engine = 32.4 hp. Brake horsepower is the indicated horsepower minus the friction of the engine.

515. Q.—How many calories of heat are developed in one explosion in a gasoline engine?

A.—How much heat is liberated by the explosion in a gasoline engine is dependent upon the ratio of air to gasoline employed, and the quantity of such mixture admitted to the cylinder.

If we represent the volume of the charge after it has been drawn into the cylinder by V_a, and the volume of the charge at atmospheric pressure by V_o, and assume "a" to be the air required per cubic foot of gas, and x the proportion of actual air to theoretical air, so that $(x-1)$ will be the percentage excess air, then the volume of cold mixture per cubic foot of gas will be $(xa+1)$, and the volume after it is drawn into the cylinder will be

$$V_a\,(xa+1) \div V_o$$

If H is the heating value per cubic foot of gas under standard conditions, the heating value of the mixture per cubic foot after it has been drawn into the cylinder will be:

$$H \div \frac{V_a}{V_o}\,(xa+1)$$

516. Q.—What per cent of the heat passes out through the outlet valve of a gasoline engine?

A.—The percentage of total heat delivered to the engine contained in the exhaust varies considerably according to the design and the operation of the particular type of unit. Results of tests conducted on a 300-hp. gasoline engine which the writer has before him indicate an exhaust loss of 24.1 per cent with the engine carrying 197 hp., 23.9 per cent when carrying 294 hp. and 24.8 per cent under a load of 335 hp.

The theoretical quantity of heat rejected may be determined by the following formula:

$$H_r = W\ C_v\,(T_t - T_a)$$

Where H_r is the heat in B.t.u. rejected at exhaust, W is the weight of gases contained in the cylinder; C_v is specific heat at constant volume; T_t is absolute terminal temperature, and T_a is absolute temperature of the atmosphere.

517. Q.—What is the pressure after explosion in the cylinder of a gasoline engine? How does this compare with compression pressure?

A.—According to Poole as reported by Kent, the maximum temperature in a gasoline engine after combustion, may be as high as 3000 deg. F. absolute, and the maximum pressure as high as 400 lb. per square inch absolute; these are high figures, however, the more usual figures being about 2300 deg. F. absolute and 250 lb. absolute.

Nothing definite can be said regarding the relative difference between the compression pressure and the pressure after combustion. This depends upon the exponent used in the pressure-volume formula, on the ratio of the original volume of the mixture in the cylinder before compression and the volume after compression, and on the absolute temperature before compression. The compression pressures may, however, range from 50 to 260 lb., depending upon the above conditions.

518. Q.—In a gasoline engine, will decreasing the clearance between the piston and head, thus increasing compression, increase the power of the machine?

A.—Reducing the clearance in a gasoline engine will increase the compression and will slightly increase the power of the engine, as it will raise the pressure after ignition. There is a point, however, beyond which this cannot be carried safely. If the compression is too high, it is likely to result in back-firing or pre-ignition, with possibly serious results.

With a heavy compression, it is usually necessary to have the ignition somewhat later than with a lighter compression, also the increase in compression and consequent increase in initial pressure after ignition may result in putting a heavier strain on the parts of the engine than it was designed for, and in this way cause rapid deterioration or even breakage of some parts of the engine.

519. Q.—How can I ascertain the correct mixture of gas with a hand analyzer attached to a gas engine? Would not the water in the muffler absorb enough CO_2 to affect the result?

At what point should I take a sample of gas on a 55-hp. twin cylinder Bruce-Macbeth engine, and why?

What would be a condition beyond improvement?

A.—A gas engine fuel mixture is correct when all of the carbon monoxide, CO, is converted into carbon

dioxide, CO_2, and all the hydrogen has united with oxygen to form water within the engine cylinder. The oxygen for this combination is derived from the air used in the mixture, and neither carbon monoxide nor hydrogen should appear in the exhaust.

If a surplus of air is used as is necessary, oxygen will show in the exhaust, in which case you are sure all the carbon monoxide has been burned, and a small loss has taken place because of the surplus oxygen and nitrogen.

If there is a scarcity of air, and as a result a scarcity of oxygen, carbon is unconsumed and smoke is the result; this is a sure indication of imperfect combustion. Secure that exhaust, therefore, that shows a slight excess of oxygen and no carbon monoxide, hydrogen or other combustible gases. The percentage of this oxygen should be reduced to the lowest limit, even to the fraction of one per cent.

The water in the muffler of a gas engine will not affect the results of the gas analysis, for this water is hot and hot water will not absorb carbon dioxide. If this water were cold it would not cause error, for the amount of carbon compound is so great that the water soon becomes saturated and then no further change takes place, the water addition being small. In the gas analyzer itself, the gas is drawn into the presence of water and little or no loss takes place.

Samples should be drawn from the exhaust pipe of the engine as close to the cylinders as possible. No error will result in throttling engines if this is not done; but in a hit or miss type, the breathing through the exhaust pipe will affect the sample if it is not taken at the place noted. The nearer the engine the better in all cases.

The percentage of carbon monoxide in the exhaust will vary widely with different gases and no stated amount will apply for all. A surplus of air as indicated by the appearance of oxygen in the analyzer is the best guide for the engine operator. This insures the conversion of all the hydrogen into water and all the carbon monoxide into dioxide, a condition which is ideal.

520. Q.—How many cubic feet of natural gas will be needed per hour by a 200-hp. engine?

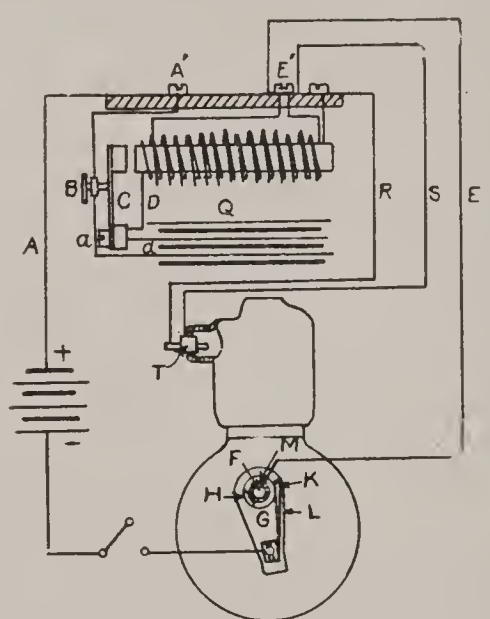
A.—A good 200-hp. gas engine will consume about

10 or 12 cu. ft. of natural gas per brake horsepower hour, assuming a heat value of 1000 B.t.u. per cubic foot of gas.

521. Q.—What are three devices used to produce ignition in the cylinder of a combustion engine?

A.—The three most common devices used to produce ignition in the cylinder of an internal combustion engine are, the high-tension or jump spark system operated by battery and induction coil, jump spark system operated by magneto, and the low-tension or make and break system operated by kick coil and battery.

In the first mentioned, we find a battery connected to the primary winding of an induction coil through the

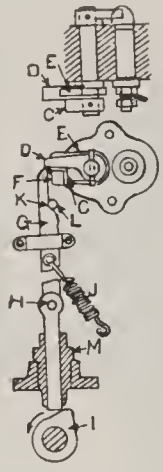


A521-1. BATTERY JUMP-SPARK SYSTEM

intermediary of a timing device H which is operated by the engine and closes the primary circuit at the required instant. To eliminate excessive sparking at the contactors, a condenser, Q, is usually connected across these points. The secondary circuit RS is connected directly to stationary spark plugs, the contact points of which are so spaced as to allow a single spark or series of sparks to jump across the permanent break thus formed.

Some systems eliminate the use of the battery and induction coil, and instead employ a magneto which derives its motion from the engine. A timing device is also used with this arrangement and the circuit between magneto and spark plug is similar to the secondary circuit of the battery-induction coil system.

The low-tension or make and break system, A521-2, ordinarily consists of a battery, switch, kick or reactance coil, and electrodes connected in series in a single circuit. If desired or required, a magneto or generator may, however, be used instead of the battery. The general arrangement is such that when the electrodes which are placed in the combustion chamber of the engine are in contact, ordinarily accomplished by the reciprocating mechanism of the engine, the current flows through the circuit until the electrodes are separated by a spring.



A521-2. MAKE AND BREAK SPARK SYSTEM

When the electrodes are suddenly parted, a spark passes between them, igniting the charge in the cylinder, if such charge is at the proper condition of mixture and compression.

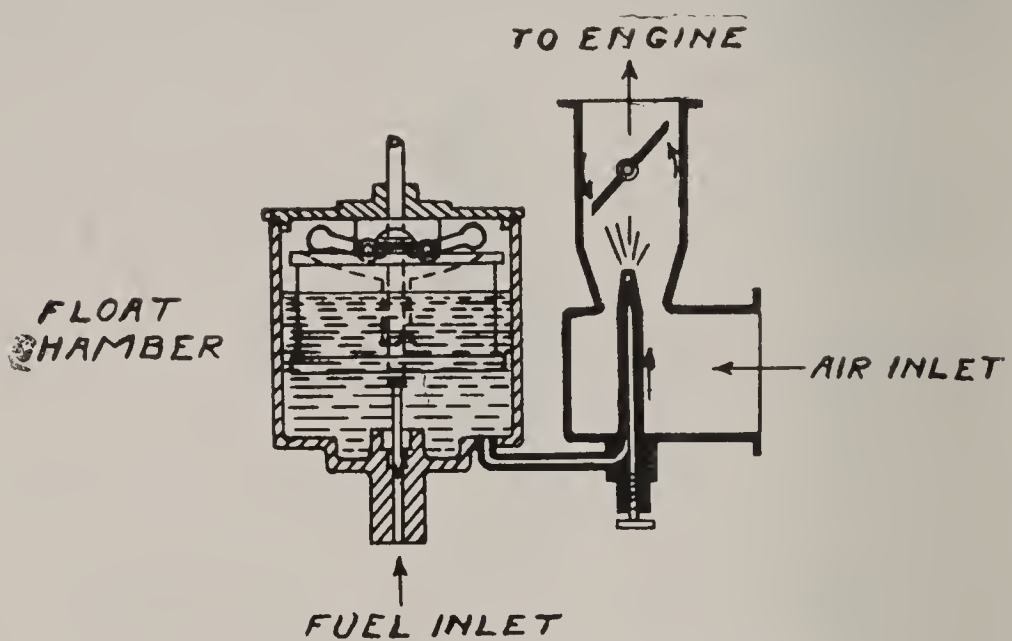
This make-and-break apparatus is comparatively simple electrically, but mechanically it is more complicated. The device used to give motion to the electrode must necessarily be more or less complicated in order to make it convenient to change the spark. It is also troublesome to prevent leakage around the moving electrode where it enters the cylinder. Lubrication is also difficult and it is hard to keep the electrode cool enough to work efficiently. For high-speed engines, special refinements are necessary in order to make the inertia effects as small as possible and to reduce the lost motion, due to wear, to a minimum.

522. Q.—WHAT IS a carburetor? Can kerosene be used in a gasoline carburetor to run a gasoline engine?

A.—A carburetor is a piece of apparatus used in connection with internal combustion engines in which

the liquid fuel used is atomized and mixed with the air which forms the remainder of the charge.

Almost any gasoline engine which has been operated long enough to have attained an average working temperature can be made to continue in operation on a kerosene and air mixture formed in a gasoline carburetor. The operation will, however, be more or less uncertain and in general it will be but a matter of a few days before the interior of the cylinder is covered with



A522. SECTION OF CARBURETOR

a thick deposit of carbon and the piston rings gummed and even burned tight in their grooves. Preheating the air and jacketing the carburetor will generally minimize the difficulty, but will not entirely overcome it.

523. Q.—Is it practical to operate two 4-cycle engines, one of 175 hp., and the other 25 hp., from the same suction gas producer, the 175-hp. engine to be located at producer and the 25-hp. 1000 ft. from it?

Would the main difficulty be in getting a sufficient supply of gas to the small engine?

Cannot this be overcome by means of an exhauster similar to or the same as used with pressure producers and connecting it in the gas main between producer and small engine at the producer end, driving it by means of a motor from generator driven by the large gas engine and by this means deliver gas to the small engine under slight pressure?

What size pipe and exhauster would be necessary?

A.—Your supposition is correct with regard to the

difficulty of getting gas to the small engine, under the conditions mentioned.

Unquestionably, this can be taken care of by using some kind of an exhauster to force a supply of gas to the small engine. Just how much capacity or pressure would be needed, is a thing that can be determined only by experiment. It would be necessary to put in a fan and variable speed motor, and work it out by trial.

The fan should be of a size to supply about 2500 cu. ft. of gas per hour, but what pressure would be required, we have no idea.

As to the size pipe, this would be the same probably that would be used if the engine were connected close to the producer, or if it might even be a little smaller, since the friction in the pipe would be taken care of by the exhauster fan, and a receiver can be placed close to the engine to take up slight variations in pressure.

524. Q.—For a 25-hp. gas engine, to avoid running the exhaust to the roof, we are considering connecting into a smoke stack from the boilers or into an 8-in. exhaust pipe from a steam engine. The 30-in. steel stack carries two 100-hp. boilers, and the 8-in. exhaust has a 90-hp. engine, 4-in. pipe, and two 1-in. pipes from boiler feed pumps. Would there be danger of explosion in the stack from unburned gases?

A.—Under no conditions should the exhaust be conducted into a smokestack, as unexploded charges of gas are liable to collect at the outlet of this exhaust pipe, which charges will be later fired by the following exploded charge. Explosions taking place from such a condition as the above, are quite likely to wreck the smokestack and it is a practice which should be condemned.

The exhaust pipe from a steam engine is generally strong enough to withstand the effect of these explosions which would be unlikely on account of the low temperature, and little trouble would arise from connecting directly into this steam exhaust pipe. Connection should be made so that the gas exhaust would enter through a Y and not be led in at right angles to the center line of the pipe.

525. Q.—What would be the cause of a gas engine's getting unduly hot (75 deg. C.) with a very light load?

It is a four-cylinder, four-cycle gas engine, ports and jackets having water forced through them by a rotary pump. It is in good condition. No water or oil passages are clogged. The viscosity of the oil is the same as is used in other engines of this type.

A.—The probable cause of heating of your engine under light load is late ignition. With everything all right, the cooling system working properly and the oil in good condition, there should be no heating except from this cause.

You do not state where the temperature of 75 deg. C. is found, but this is not a very high temperature for a gas engine, unless you mean for the cooling water coming from the jacket; in that case, the temperature is somewhat high, but in any case we believe that advancing the spark will remedy your difficulty.

526. Q.—How would you suggest muffling the exhaust of an oil engine?

We have a 4-in. line exhausting into the open and I am desirous of knowing whether by building a concrete cistern with a 12-in. stack I could eliminate some of the noise now taking place without a loss of power.

A.—The concrete tank which you propose is probably as good a method as any of muffling the exhaust from your oil engine. The principle of muffling such an exhaust is the attaching of some device to the end of the exhaust pipe, which permits a gradual expansion of the gases instead of the sudden expansion which comes when the gases are released at once from the end of the pipe.

Attaching a chamber to the end of the pipe, such as a piece of 10-in. pipe a couple of feet long, and providing a 4-in. outlet from the end of this, will reduce the noise considerably; but if this device be placed inside the concrete pit, and a 12-in. stack provided for carrying away the gases, it will work even better. Such a device would add very little to the back pressure.

527. Q.—How is the charge ignited in the Diesel engine?

A.—In the Diesel engine, the air is compressed to a predetermined temperature above the firing point of the fuel, which is then blown in as a cloud of vapor, and

ignites spontaneously without explosion solely by reason of the heat of the air generated by the compression.

528. Q.—What are the relative economies of a coal fired steam plant and an oil engine for 50 hp. used 18 hr. out of 24? Coal costs $3 a ton and oil 5 cents a gallon.

A.—The following example will give you some idea of the comparison:

A 50-hp. noncondensing engine will use about 40 lb. of steam per indicated horsepower per hour, or

$50 \times 40 = 2000$ lb. of steam per hour.

One pound of coal will evaporate about 7.5 lb. of water per hour; then,

$2000 \div 7.5 = 266$ lb. of coal required per hour;

$266 \times 300 \div 2000 = 40$ cents, the cost of coal per hour to operate the steam engine.

The manufacturers of oil engines guarantee to deliver one horsepower on less than one pound of oil, or say 0.8 lb. Then, $50 \times 0.8 \times 5 \div 8.3 = 24$ cents per hour fuel cost for an oil engine.

529. Q.—Would it be practicable to install a small electric lighting and refrigerating plant to be operated by means of a gasoline engine? This would be for use in a country store.

A.—It would be entirely practical to drive a small electric light and refrigerating plant with a gasoline engine. The only question in regard to this would be whether it would pay. Gasoline is rather expensive for fuel, and while for small amounts of power, in a place where electric current is not available, it may be the most practical method, there is always a question whether the return will pay for the investment and operating cost.

The gasoline engine, under this kind of a load, will run with very little attention, but it will take approximately a pint of gasoline per hour per horsepower.

It is probable that, with this kind of a load, you might find it advisable to use a kerosene or a heavy oil engine rather than gasoline. Kerosene does not work particularly well where there is a load that varies widely; but for refrigeration and electric lighting, this would not be the case, and we believe an oil outfit could be found which would work very well.

PART V

Electrical Equipment

Direct Current Generators

530. Q.—In what classes may direct current dynamos be divided according to method of excitation?

A.—Direct current dynamos may have their fields excited by field coils connected in series, shunt or a combination of series and shunt called compound windings.

531. Q.—Why is the series-field shunt used on a compound-wound dynamo, and would the machine generate without one?

A.—In the design of compound-wound, direct-current generators, it is impossible to foretell exactly how the generator will compound when a load is placed upon it, due to the influences which are not under the control of the designer or builder of the machine. It is, therefore, the custom to make the series winding of the fields greater than is necessary to hold the voltage of the generator up to its normal rating when fully or overloaded. It becomes necessary, therefore, to place a resistance shunt across the terminals of the series winding, so that it will exactly compensate for the load upon the generator. This shunt is usually of German silver, and is put upon the generator after a test is made in the factory, when the exact resistance required to compound the generators properly can be computed. You will see, therefore, that the generator would generate current without the shunt, but more than likely, the machine would be considerably over-compounded; that is, would give a higher voltage reading at full load than at light load.

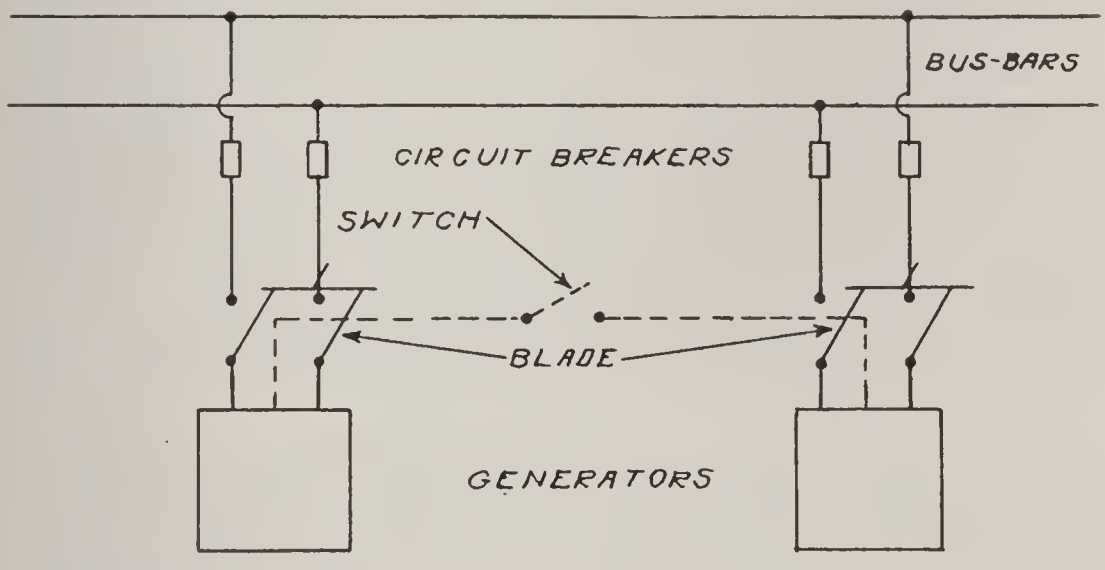
532. Q.—What horsepower is needed to drive a 35.7-kw. generator?

A.—An approximate rule is that the indicated horsepower of an engine should be $1\frac{1}{2}$ times the kilowatt capacity of the generator, or, in your case, $35.7 \times 1\frac{1}{2}$ would give 53.5 indicated horsepower.

533. Q.—Two 1200-volt d.c. compound-wound railway generators are driven by synchronous motors and

operate in parallel on the d.c. end, as shown in A533, through one breaker switch and one blade switch for each machine. Could an equalizer be connected as shown dotted on a pedestal? Or would it be better to put in two pedestals and switches? If the breaker on one machine should open, would it be reversed by the other, the equalizer connection being closed?

A.—While it would be entirely practicable for you to have but one equalizer switch between your two direct-current machines, it would be better to have one switch for each of the machines, that is, as you un-



A533. CONNECTION OF GENERATORS IN PARALLEL

doubtedly have a pedestal for each generator, we would suggest mounting a separate equalizer switch on each of these pedestals.

Should the circuit breaker open on one of these machines no reversal will occur. In fact, generators connected in parallel and tied together with equalizers, are safeguarded against reversal due to the opening of a circuit breaker on any one of the machines.

534. Q.—WHAT OPERATIONS are necessary to throw a d.c. compound-wound generator in parallel with others already in service?

A.—Consider that you have one compound-wound generator in operation, with all switches and circuit breakers on the switchboard in, including the equalizer switch. Having the incoming machine up to speed, adjust the voltage so that it is approximately equal to that already being carried on the bus bars. Having done so, throw in the equalizer switch on the incoming ma-

chine. Then, as a special precaution, again compare your voltages, and if they are about alike, you may throw in the incoming machine. In the case of direct-current compound-wound generators, the load can then be proportioned between the two machines by means of their respective field rheostats.

535. Q.—What is the right way to shut down one of two direct-current generators running in parallel?

A.—The proper way to shut down one of two machines in parallel is, first, to cut resistance into the rheostat, in the field circuit, which will diminish the voltage of the machine to be shut down until it is carrying practically no current. This will be indicated by the ammeter reading. When you have reached this point where the ammeter on that machine reads practically zero, pull the machine switch connecting this machine to the switchboard and the equalizer switch. You can then turn in all the resistance in the rheostat and cut the voltage down to practically zero, then shut down the engine.

If you pull out the dynamo switch when the generator is carrying a large load, you will get a heavy arc across the terminals of the switch, which will be likely to burn the switch blades and clips, but do not throw in all the resistance in the rheostat and run the voltage down even lower than that of the other dynamo, because when you do this, you will have the machine whose voltage has been cut down working as a motor off the other machinee.

536. Q.—What is the cause of a dynamo burning out?

A.—The cause of a dynamo burning out is always because it gets too hot. When we say that it burns out, we mean usually that the insulation is burned, not the wires, and it comes from the passage of too large a current. This may be due to wear on the insulation which permits a short circuit inside the winding, or it may be due to an overload on the machine.

537. Q.—Number 1 generator is 40 kw. direct connected, and No. 2 is 30 kw. belt driven. At noon we cut out No. 2 and cleaned the commutator. When we cut it in again at 4:30, it was reversed. There is an

equalizer switch, and both are compound wound with carbon brushes. The voltage was brought up equal to No. 1, but on closing the switch, there was a flash and No. 2 reversed.

A.—The most likely cause for this reversal was pulling the field switch before the voltage of the machine had been reduced to zero by turning in resistance in the rheostat. If this were done, the self-induction flash-over of the fields was liable to produce reversal of polarity.

This effect is not uncommon, and is the most probable cause in your case. The reversal could not have been caused by the compound coil, since that tends to keep the polarity in the same direction as the shunt coil; in fact, it would probably be possible to bring your machine back to its original polarity by closing the compound coil connections to the equalizer bus and to the main bus. Of course, the other side of the main switch must not be closed when this is done, or you will throw the full voltage onto the armature and cause trouble.

The only thing which can cause reversal of polarity in a generator is reversal of the direction of magnetism in the poles, and this is caused by change in the direction that the shunt current flows around the field. If the residual magnetism is changed as the machine shuts down, then as it starts up its polarity will be reversed and current will be sent through the field coils in the opposite direction.

Of course, when your machine reversed, your shunt coils and compound coils were opposing each other, because of the action through the equalizer bus, and you had a differential winding instead of a compound winding, so that the series coil tended to cut down voltage instead of raise it.

538. Q.—I AM in charge of a direct-current generator which operates at a speed of 1200 r.p.m. and which is rated at 15 kw., 160 v. This machine fails to generate at times and I find much difficulty in making it build up its voltage.

A.—Evidently the machine you refer to is suffering from a case of chronic lack of residual magnetism. We would suggest that you disconnect the field coils from the armature and with the individual coils in series pass

a full strength field current through them, letting the flow continue for a period of about one minute. Then gradually reduce the flow until it is practically zero and disconnect the source of excitation. Preferably this should be done with the machine at rest.

539. Q.—What is the best way to re-establish the residual magnetism in the field when this disappears? If I do this by passing current from another generator or a storage battery through the field coils, must this be done while the machine is running?

A.—Preferably this should be done with the machine at rest. You may then re-connect the machine and bring it up to speed, after which you should experience no further trouble. There is a possibility, however, that the flow of the exciting current was in the wrong direction, in which event the machine will not produce the correct polarity; that is, the negative lead will appear positive and the positive lead negative. This may be corrected by again passing the exciting current through the coils, but in the opposite direction.

540. Q.—For two 110-v., 450-amp., direct-current generators operating in series, what number of amperes can these machines deliver at full load. Is it 450 or 900 amp.?

There is a separate ammeter for each machine and these indicate 350 amp. each when the generators are operating almost fully loaded. Does this mean they are delivering 700 amp.?

A.—If your two direct-current generators are each rated at 450 amp., 110-v., and are connected in series, the voltage delivered by these machines will be equal to 110 plus 110, or 220; or, in other words, the sum of the voltages of the individual machines. The current capable of being delivered by these machines is equal, however, only to the current capable of being delivered by either one of the machines, or, in your case, is equal to 450 amp. at full load.

Even though each one of these machines is provided with its individual ammeter, if each ammeter indicates a flow of 350 amp., the current delivered by both of these machines is equal to 350 amp., and not equal to 700 amp.

541. Q.—We have a 125-v. 220-amp. d.c. generator. Since the machine was installed there has been sparking and chattering at the brushes, and the machine has spells of whistling. How can we remedy these troubles?

A.—It is a difficult proposition to suggest what may be the difficulty with a sparking commutator. Causes for this are: Improper setting of the brushes, rough commutator, high mica, or an overloaded machine. Just which one of these may cause the trouble is almost impossible to say without a careful examination of the machine in operation.

The chattering of the brushes would seem to indicate high mica—that is, the mica projecting above the commutator bars. This can be remedied by undercutting the mica.

See that the brushes are set exactly as far apart around the commutator as the poles—that is, if you have a 4-pole machine, the brushes should be exactly ¼ of the circumference apart; if a 6-pole machine, 1/6 of the circumference, and so on. If the brush holder is adjustable, try moving the brush holder ring backward and forward a little to see if you can lessen the sparking. Hold the point of a lead pencil against the commutator as it revolves to see whether there are apparently high spots; either mica or one or two bars that are higher than the rest of the surface.

542. Q.—We have a 17-kw. d.c. compound-wound direct-connected dynamo that has developed a tendency to spark at the brushes. Eleven or 12 segments on the commutator will be burned and blackened, and also a little flattened, and directly opposite these segments about the same number of segments will be in the same condition. What is the cause and the remedy?

A.—The blackening and flattening off of your commutator bars is probably caused by the sparking, and naturally this would occur on opposite segments, since the sparking will tend to occur on both sets of brushes at the same time.

Just what causes the sparking is difficult to state without a careful examination of your machine. From the fact that it occurs at certain points on the commutator, the indication would be that there is something wrong either with the commutator at these points, or else with the armature coils connected to the commutator

bars affected. The sparking may be due to loose connection between the coils and the commutator bars, to an open coil or a short circuited coil, or to high mica between commutator bars. By holding your thumb nail against the commutator as it is revolving, you can easily detect the high mica if it exists. If there is an open coil, there will be a sharp flash when the bar connected to that coil passes under the brush. A loose connection would mean simply a high resistance in the coil, and would cause a flash or spark, but not necessarily of great violence. A short circuited coil would be apt to show a little jerk in the voltage as the brushes pass the bars connected to that coil.

543. Q.—What is the cause of the pitting or burning out of mica between commutator segments at or near the end of the commutator? The leads to the segments are not loose and the burning occurs in rings from 0.125 in. to 1 in. wide. The mica is charred and the pitting sometimes extends down to the clamping ring where the segments finally become grounded. At the time of starting the motor a ring of fire encircles the commutator but usually is extinguished when the motor reaches full speed. What compound can be employed to fill up the spaces between the segments?

A.—It is probable from the location of the pitting that the trouble is due to oil from the shaft bearing collecting on the end of the commutator, ensnaring carbon dust from the brushes, and thus bridging the segments. The short-circuit current, together with the peripheral speed of the commutator, probably rids the mica spacers of the carbon dust after the motor has operated a while. After cleaning out the pitted mica with a metal file the space can be filled in with a mixture of powdered mica with shellac or an insulating varnish. To avoid any likelihood of this trouble recurring it will be advisable to examine the bearing for escaping oil, and provide means for collecting or eliminating such oil as may escape.

544. Q.—What is armature reaction?

A.—Armature reaction is the result of a magnetic field set up by the armature windings. It may be either a cross field at right angles to the main field set up by the field windings, or with the brushes shifted to avoid

sparking, will be partly cross field, and partly negative field opposed in direction to the main field. The greater the amount of the armature current, the greater will be this reaction, also, the greater the shifting of the brushes from a line at right angles to the magnetic direction of the main field, the less will be the cross field and the greater the negative field.

545. Q.—How can the commutator of a small exciter be trued up when no lathe is available?

A.—Satisfactory truing up of the commutator of a small machine can hardly be accomplished without placing the armature in a lathe. Should, however, the commutator be off true but a trifle, a skilled machinist or mechanic may be able to remedy the trouble by going over the surface of the commutator with a smooth file, the armature revolving at a speed of about 200 r.p.m. Best results can be obtained only by placing the armature in a lathe.

546. Q.—A 4½-kw. shunt bipolar motor has the magnets wound for 110 v. with No. 26 B. & S. gage wire, 5 lb. to each spool. One spool has burned out. Using No. 27 wire, how much will it take to balance the other spool?

A.—When unwinding the damaged coil of your motor, carefully count the number of turns of wire thereon, so that you may be able to use the same number of turns when rewinding it.

Theoretically, it would not be permissible to use a No. 27 wire, as you suggest doing, but practically it can very easily be done without any effect whatever on the satisfactory operation of the motor.

547. Q.—How can the back pitch of an armature winding be found?

A.—A formula developed by Arnold, which is applicable to all cases of armature winding, gives a convenient means of determining the pitch. This formula is,

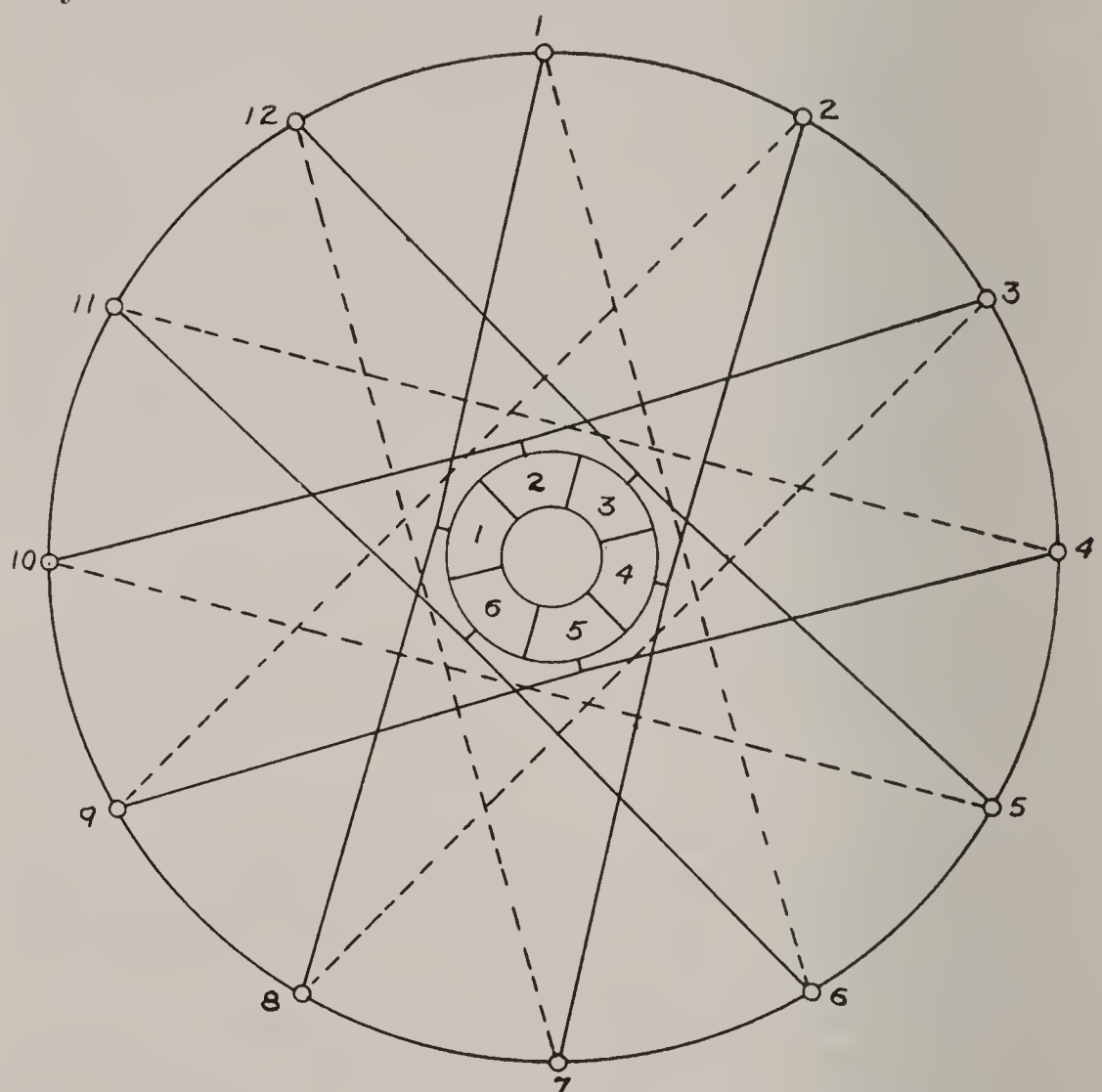
$$y = \frac{2}{n}\left(\frac{z}{b} \pm a\right)$$

Where y = pitch;

z = total number of conductors;

b = number of conductors in one element of winding;
a = a constant;
n = number of poles.

To understand the application of this formula thoroughly, some explanation of the various factors is necessary.



A547. ARMATURE WINDING DIAGRAM

In determining the value of z, we ignore the actual total number of conductors and consider each coil as a single turn around the armature, each side of a coil counting as one conductor. An element of the winding is that portion of the winding between two commutator bars. The term b, therefore, represents the number of conductors between two commutator segments. In this case, as in the case of the total number of conductors, we consider each side of a coil as one conductor. The number of poles is denoted by the term n. In a bipolar machine, this would be equal to 2. In a 4-pole machine, n would be equal to 4, etc.

The value of a for multipolar machines determines the number of branchings of the circuit through the armature. If a is equal to one, there will be a single branching, or 2 circuits in parallel. If a is equal to 2, there will be 2 branchings, or 4 circuits in parallel, etc.

The term y denotes the pitch, or spacing, or the number of conductors passed over in connecting together 2 conductors. The term y can be better understood by referring to Fig. A547, which represents the end view of an armature with 12 conductors and 12 winding spaces. If the value of y is taken as 5, and we wish to know to what conductor 1 should be connected, we begin to count with conductor 2, and count 5 conductors, which would bring us to conductor 6, this being the fifth conductor, or the fifth winding space from conductor 1. Conductor 3 would connect to conductor 8. If the value of y is 7, conductor 1 would connect to conductor 8, conductor 3 to conductor 10, etc.

To determine the number of conductors, if we have a given pitch, we transpose the terms of the formula, which gives us,

$$z = b \left(y \frac{n}{2} \pm a\right)$$

Alternating Current Generators

548. Q.—Kindly give me a formula by means of which I can calculate the number of poles required for any alternating current generator, showing the frequency per second and the speed.

A.—Consider first a two-pole machine. With each revolution of the armature one cycle is completed so that if there are n revolutions of the armature made per second, there will be n cycles per second. If, however, the number of pairs of poles be p, then the frequency f will be equal to the product of the number of pairs of poles by the number of revolutions per second, or $f = p\ n$.

Transposing and solving for p we have $p = f \div n$ or the number of pairs of poles equals the frequency in cycles per second divided by the number of revolutions of the armature per second.

549. Q.—If a dynamo has a capacity of 500 kv.a. how is it different from one of 500 kw.?

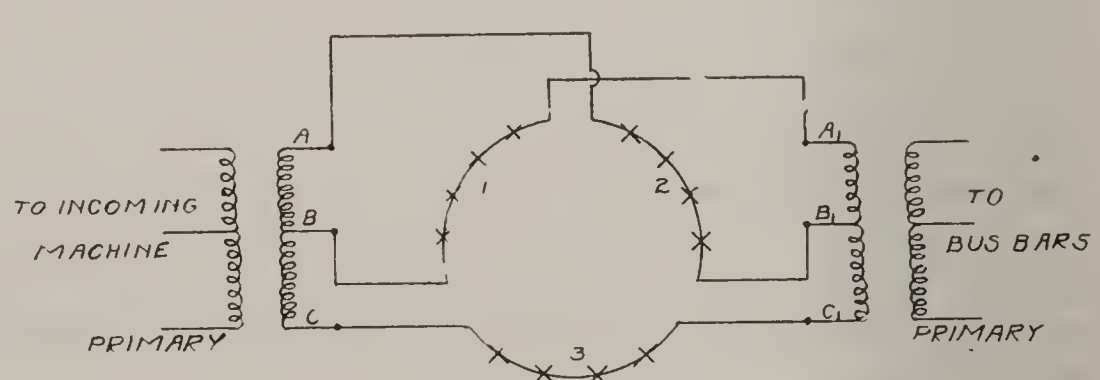
A.—A dynamo having a capacity of 500 kv.a. might or might not develop the same power as one having 500 kw. capacity. "Kv.a." means kilovolt amperes.

Due to self-induction in an alternating current circuit, the current lags behind the applied electromotive force. If the electromotive force and current are in phase with each other, as for direct current, the power developed in watts is the product of the voltmeter and ammeter readings; while when out of phase, this product will be greater than the true watts, and is known as the "apparent watts." The ratio of the true watts to the apparent watts is called the "power factor."

Thus, the kv.a. is the product of the voltmeter and ammeter readings divided by 1000, and to get the true kilowatts, this must be multiplied by the power factor, which varies from as low as 0.70 in the case of the circuit on which there are induction motors, to as high as 0.95 in the circuit on which there are incandescent lights only.

The purpose of the kv.a. rating is to protect the machine, as the tendency might be to raise the current, so as to bring the load up to full rating in kilowatts. With a low power factor, this would, of course, result in excessive current and heating.

550. Q.—Two 3-phase, 2300-v., 60-cycle alternators are to be run in parallel. One is 500 kw., the other 350 kw. Leads are not marked as to phase. How can they be tested for phase so as to connect up properly?



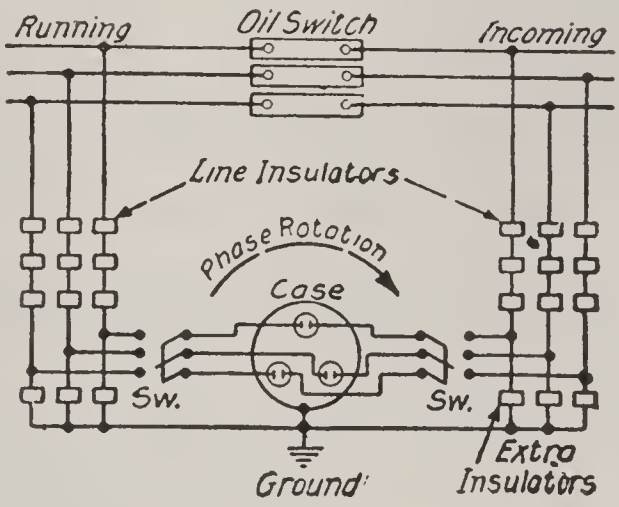
A550-1. TRANSFORMER CONNECTIONS FOR PHASING OUT

A.—Two banks of transformers open delta-connected on the primaries may be arranged as shown in A550-1, and connected as indicated. Transformer secondaries are then interconnected at random through lamp banks,

and with the incoming machine up to speed and voltage, correct connections are indicated by one bank of lamps being dark and the other two about up to candlepower. If all three banks are bright in rotation, so that the light appears to travel around the circle, transpose any two machine terminal connections to obtain the desired result.

Lamp banks should consist of at least two lamps each for voltage corresponding to that across a transformer secondary.

If available, an electric glow type of synchronism indicator may be used without transformers. This con-



A550-2. GLOWER CONNECTIONS FOR PHASE TESTING

sists, as in Fig. A550-2 of a case with three electrostatic glowers connected to the lines by disconnect switches and line insulators. With the machine under test up to speed and voltage, switches are closed and the glower elements will flicker, somewhat like synchronizing lamps. The apparent direction of rotation of brightness indicates relative speeds of the line and the tested machine. At synchronism, rotation of the flicker will stop; the upper glower, whose terminals are connected to the same phase, will be dark, and the other two will be steadily lighted at half brilliance. If this condition cannot be obtained, transpose any two connections of the incoming machine and test until synchronism is secured.

551. Q.—WILL YOU give some information regarding the construction and operation of the Warren inductor alternator? Could two such machines rated at 60-kw., 1100-v., 60-cycle, two-phase each, be operated in parallel with any degree of satisfaction?

A.—In an inductor alternating-current generator, the armature generally consists of a stationary laminated ring, the winding being embedded in slots in its inner surface. This ring is fixed inside a cylinder frame or yoke, which, with the revolving element that rotates inside the ring constitutes the magnetic circuit.

The armature is built up of annealed sheet steel, the sheets being punched in segments which dovetail into projections in the yoke casting. Ventilating ducts are provided at suitable intervals in the core. The armature winding is composed of machine-wound coils, held in place by wedges. It is connected by insulated leads to the terminal block of the machine, which is mounted on one side of the yoke.

The revolving element carries no winding whatever and the machine has no sliding contacts. The spider or hub of the keeper is cast steel, the pole pieces being built up of laminated steel and cast with it into one piece.

The field winding consists of one large stationary coil, encircling the keeper and supported by the outer frame or yoke. This coil is separately excited with direct current. Being thus all magnetized by one coil, the pole pieces are all of the same polarity, and as the keeper is rotated the magnetic lines are inserted and withdrawn from the armature coils each time a pole passes. These machines can be wound to generate either two or three-phase current.

Alternators of this type may be operated in parallel, but from the experience of the writer the results have never been very satisfactory.

552. Q.—A 150-KW. alternator is to be floated on the line as a synchronous motor, to improve the power factor. To build up the field enough to be of benefit will overtax the exciter capacity. The alternator can be run as a generator self-exciting, but can it be run as a motor under self-excitation with benefit to the power factor?

A.—We are not quite clear whether you wish to run your generator taking excitation from its own armature or from an exciter which is driven from the shaft of the generator.

If the former is the case, you could not work it as a synchronous motor with this kind of excitation. If

the latter is intended, we see no reason why you should not be able to get sufficient excitation from the exciter furnished with the generator to give you a leading current, when using the generator as a synchronous motor.

Of course, you would have to start the motor from some outside source. This could be started by driving the exciter as a direct-current motor, starting the synchronous motor unloaded, getting it into synchronism and throwing it on the line, and then changing the connections so that the exciter would take up its normal function.

Of course, whether you can get sufficient amount of excitation to correct for your power factor would depend on your local conditions and how much you have to advance the current in the synchronous motor.

553. Q.—What are the connections for a 2300-v. 3-phase revolving field generator, belted type, to be used as a generator for parallel operation, also as a synchronous condenser on the outgoing line?

A.—We presume that what you desire is at times to operate one of your three-phase alternating-current generators as a synchronous motor.

Under such condition, it is unnecessary to change the wiring. In order to operate this machine as a condenser, bring it up to speed and synchronize it as you would for parallel operation. Then, after it is in step with the other machines, disconnect from the source of power supply driving it.

Due to the fact that your machine is belt-driven, and we assume by either a gas or steam engine, we are afraid you will have some difficulty in disconnecting the generator from its driving unit. We would suggest that you install a small motor, providing this with a small belt, which you may throw onto the pulley of your machine in place of the larger one which ordinarily drives it. Then, when you want to run this generator as a condenser or synchronous motor, take the large belt off, replacing it with the smaller one. Start up the small driving motor, letting it bring the condenser up to speed, then synchronize as for parallel operation.

After the machine is in step, you may throw the small belt off and stop the small motor. This, however, is a somewhat cumbersome procedure, and it might be

advisable even to go to the expense of using some form of direct gearing between the small motor and the condenser, mounting the small motor in such a manner that it may be moved a short distance after the condenser is thrown onto the line, thus allowing the gears to be out of mesh. The action of the condenser can then be controlled by regulating its field current.

Motors

554. Q.—What indicates that the shunt and series field windings on a motor are opposed?

A.—If the shunt and series windings on a compound-wound motor are opposed, this fact will become evident by an increase of speed as the load on the motor increases.

555. Q.—If the field circuit of a motor breaks, what will happen?

A.—Generally, if the field of a direct-current motor be opened, the motor will stop. There is a possibility, however, in the case of a compound-wound motor, that should the shunt field open, the motor will continue to operate, due to the series field, but at an excessive speed.

556. Q.—If a motor runs at 1000 r.p.m. and the field strength is increased 25 per cent, how fast will the motor run?

A.—The change in speed of a motor with field strength depends on the characteristics of the motor, but in a rough way, the speed varies inversely as the field strength, so that if you increase the field 25 per cent, you cut down the speed 25 per cent, or from 1000 to 750 r.p.m. This would not be accurately true, however, in the case of a loaded motor, since the increase of the field strength increases the torque, and this would tend to speed up the machine, with the load remaining constant.

557. Q.—How do you figure the speed of a direct-current motor?

A.—The speed of any direct-current motor is equal to:

$$\text{R.p.m.} = \frac{E\ 10^8\ 60b}{2p\ F\ n}$$

Where E = counter-electromotive force.
b = the number of circuits in parallel in the armature winding.
p = number of pairs of poles.
F = the flux entering or leaving the armature per pole.
n = total number of conductors on the armature.

558. Q.—When running at full rated capacity, how many kilowatts per hour would a 35-hp. motor require? How would you in general calculate the horsepower of any induction motor.

If a motor runs at less than full load, will it use less current? Would a variable speed motor running with reduced load operate with better efficiency than a constant speed motor under like conditions?

A.—A horsepower is equivalent to 746 w., or 0.746 kw. For 35 hp. we would, therefore, have 35 times 0.746, or 26.1 kw. This would give in one hour 26.1 kw.-hr., but you must remember that the kilowatt, like the horsepower, is a measure of the rate of doing work, not a measure of energy. A kilowatt-hour is a measure of energy expended.

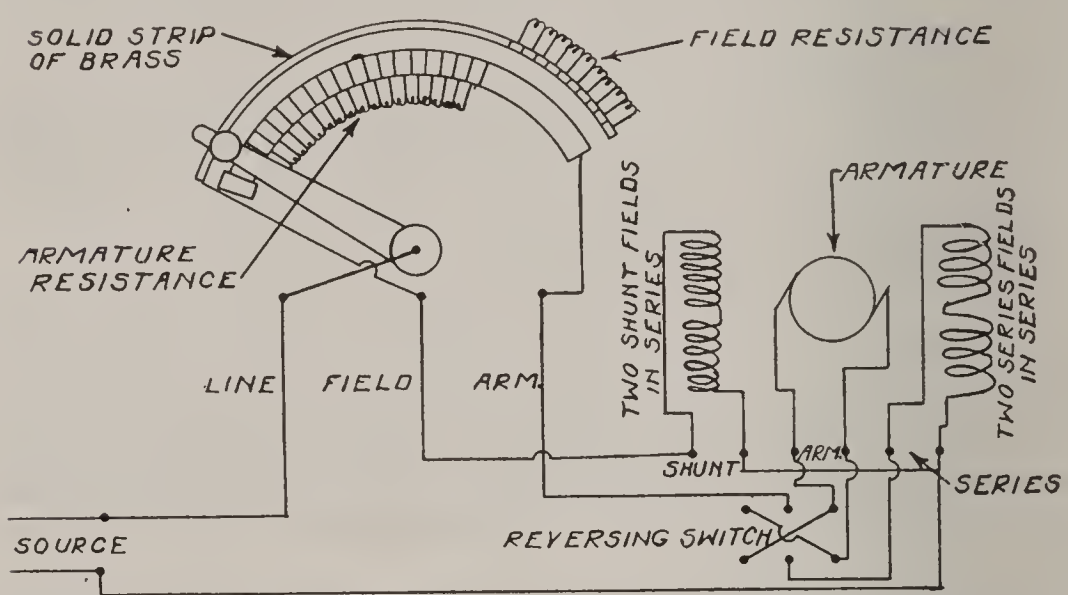
The value 26.1 would be the kilowatt equivalent of the output of the motor, and the current consumption would be larger, depending on the efficiency of the motor.

In regard to figuring the horsepower of an induction motor, if you have the measurement of the kilowatts input, divide this by 0.746, and it will give you the horsepower input. Then, the horsepower output would depend on the efficiency, which for a 35-hp. induction motor would probably be about 85 per cent, depending somewhat on the power factor. Any motor running at less than full load takes less current than if running full load. The current drawn is directly in proportion to the load carried.

There is no benefit in the variable speed motor over the constant speed, so far as efficiency is concerned. It is simply a matter of convenience of operation. As a matter of fact, in order to get a variable speed alternating-current motor, it is usually necessary to throw in some external resistance or induction, which will cut down the efficiency of operation.

559. Q.—Please give a diagram of connections for a d.c., 220-v. compound-wound motor, controlled by a Cutler-Hammer starting and controlling box. Box terminals are marked "line," "armature," "shunt field." Reversal of motor is by means of a double-pole, double-throw knife switch.

A.—Compound type controlling rheostats operate on the principle of first increasing speed from below normal (as rheostat is moved from left to right) to normal speed, and finally increasing above normal by weaken-



A559. CONNECTIONS OF COMPOUND-WOUND MOTOR AND STARTER

ing the field. Fig. A559 gives a diagram of the connections. There is no resistance introduced into the field circuit until all resistance is cut out of the armature circuit. The reversing switch reverses the current in the armature only.

560. Q.—Can a shunt-wound motor be connected direct to line in case the starting box is out of order?

A.—The proper procedure is to provide a starting box whereby the current flow through the armature windings when the motor is first connected to the supply lines is cut down to a minimum value. Then as the motor increases in speed, more resistance is cut out of circuit until rated speed is reached, when all of the resistance may be cut out of the armature circuit.

With some forms of starting boxes, connections are so made that as the resistance is cut out of the armature circuit, it is cut into the shunt field circuit. As a con-

sequence, the maximum amount of current is allowed to flow through the shunt field while the minimum amount passes through the armature circuit. With manipulation of the starting box, however, the resistance, as already stated, is gradually cut out of the armature circuit and into the field circuit. This action cuts down the flow of current through the shunt field and materially assists in increasing the speed of the motor.

If the starter is not available, the motor may be started by connecting the field direct to line, then using a water rheostat in series with the armature to bring the motor up to speed. When at full speed, short circuit the rheostat.

561. Q.—What makes a series motor run away when the load is removed?

A.—In a series motor, armature and field are connected in series, and both carry the same current. The counter-electromotive force depends on the strength of the field and the speed of the armature. If the load is removed, little current is drawn, the field strength is reduced tremendously, and the motor speeds up in the effort to create a proper counter-electromotive force.

562. Q.—Why is the series motor used for electric traction and the shunt motor for shop drive?

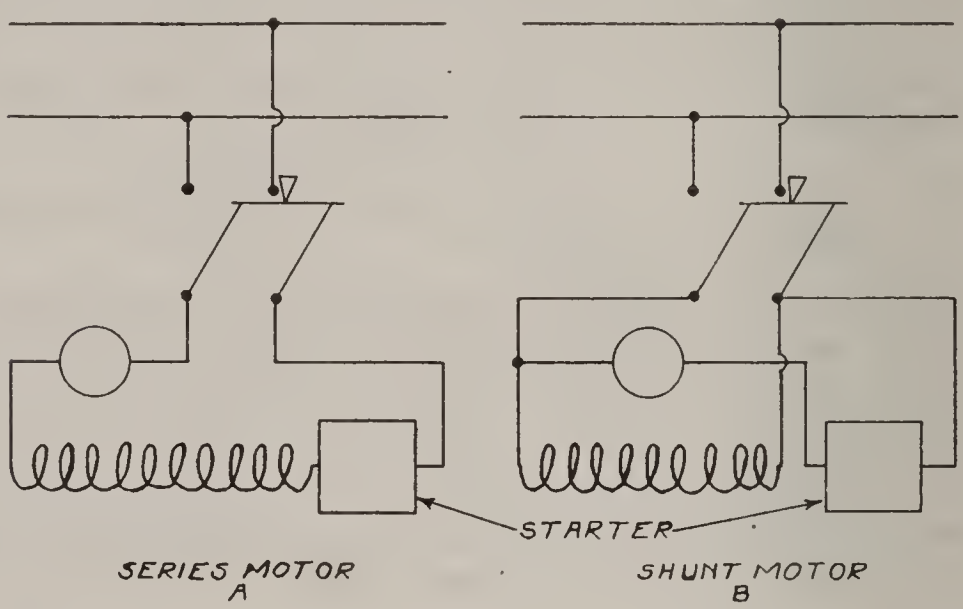
A.—For electric traction, we want a motor whose speed can be varied over a wide range. This calls for the series motor. For shop drive we want a motor whose speed will be fairly constant over a wide range of load, and the shunt motor furnishes this condition.

563. Q.—What changes are necessary to change a 110-v. direct-current generator, belt driven, into a motor? Would it require some form of resistance in starting? Would a bank of lights do?

A.—The changes necessary in order that a direct current generator may be run as a motor, consist in shifting the brushes back from their position when the machine is used as a generator, so as to give negative lead.

It is always advisable to have a resistance in series when starting up a d.c. motor. For a series-wound motor, running from a constant voltage circuit, this will be placed in series with the motor windings. For a

shunt-wound motor the starting resistance should be in series with the armature winding, but the field winding should be thrown directly across the mains. The resistance in either case should have a current carrying capacity at least 1.5 times full load armature current. A bank of lamps in parallel can hardly be used, except for very small motors, as too many lamps will be needed to allow the required current to pass, and they cut down voltage too much. A wire resistance or a liquid rheostat will be better.



A563. STARTER CONNECTIONS FOR D.C. MOTORS

564. Q.—PLEASE GIVE wiring diagram for a Sprague type 2-phase induction motor with squirrel-cage armature. The service is 2-phase, 3-wire.

A.—Figure A564 shows a diagram of connections of a 3-wire, 2-phase induction motor. In this sketch, we have shown the 3-line wires and the lines leading to the motor. The switch in this drawing is shown in the down, or starting, position. After the motor has reached its full speed, the switch is thrown up to its running position, thereby cutting out the auto-converter. By this arrangement, the motor can be started at a distance.

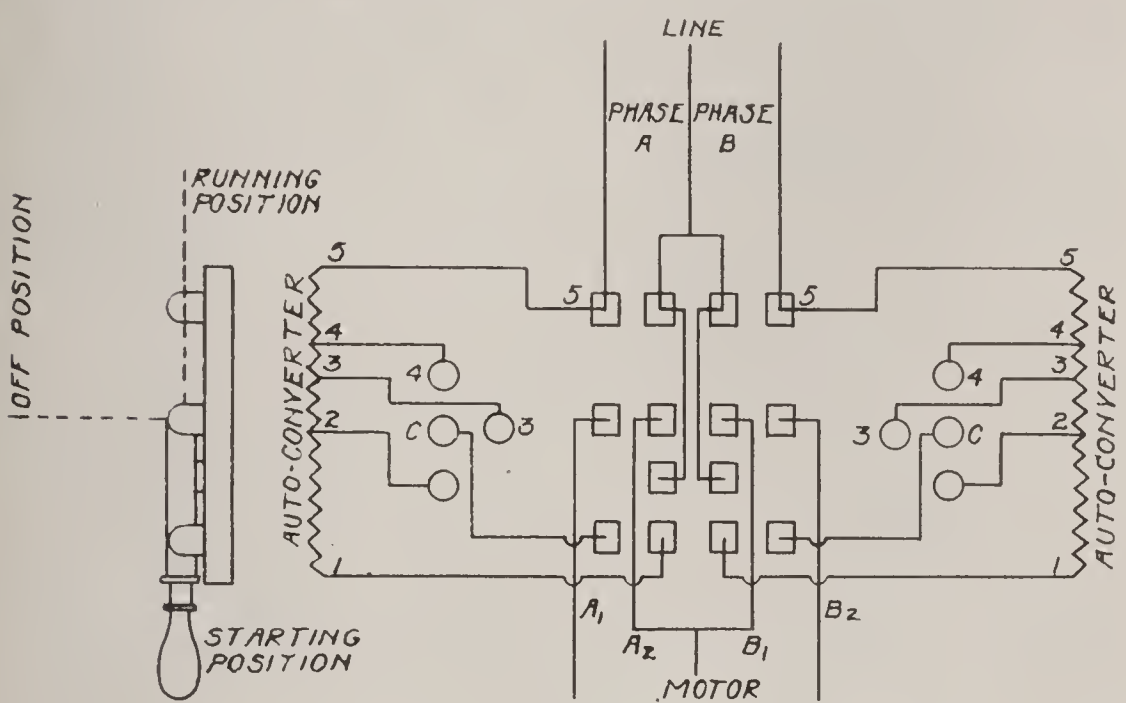
Connections can be made from C to either 2, 3 or 4, giving 3 different starting electromotive forces and starting torques. The maximum electromotive force and torque are obtained by connecting C and 4. For minimum electromotive force, C and 2 are connected.

565. Q.—I HAVE a Westinghouse textile type induction motor rated at 2 hp., 550 v., 3-phase, 60 cycles, 2.16

amp., 1725 r.p.m., 48 slots which must be rewound, and I would like to learn how to go about the job.

The coils are wound with about a number 18 wire.

A.—First count the span of one coil as you take out a whole coil, meaning the number of slots that one coil spans over. For instance, one side of your coil lies in slot No. 1, the other side lies in slot No. 10 or No. 11; counting the slot where the first side of your coil is placed, you go 1-2-3-4-5, etc., until you come to the slot where the other side of this same coil is placed.



A564. CONNECTIONS FOR SQUIRREL-CAGE INDUCTION MOTOR

In Fig. A565-1 is shown what is meant by the span or pitch of a coil.

Next count the number of turns in your coil and determine size of wire, whether single or double cotton covered. Alternating current motors are usually wound with double cotton or cotton enamel. Having determined these, you take at least one coil out of your slot and form this coil as near as possible to its normal size as it lies in the slot, or you can measure the lap of the coil over the lamination of your motor, the width of the span and the shoulder of the coil; by this is meant the distance of your sides lying in the slot from bend to bend.

This done, you count the number of coils per group or per pole per phase. If your motor runs at 1725 r.p.m.

at 60 cycles, you have a 4-pole motor and being a 3-phase motor, you multiply 3 by 4 = 12 and divide the number of coils, which is 48, by 12 = 4 and you get the number of coils per pole per phase.

Next determine the form of connection whether star or delta. In your case, it is more likely a star connection. This can be ascertained by looking at your leads or jumpers from group to group; if you see three wires connected together, it is a star, but if you find on your

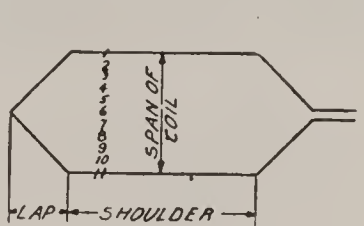


FIG 1

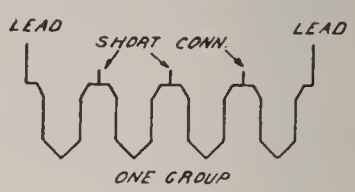


FIG 2

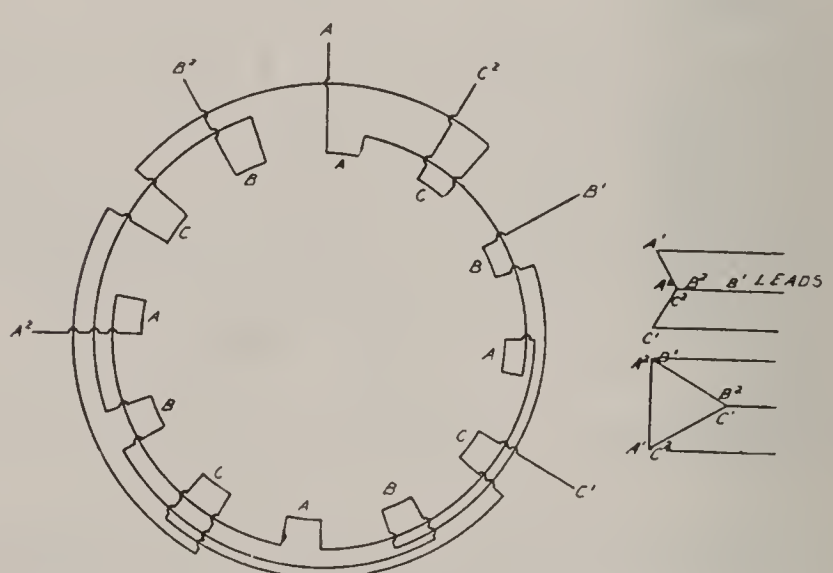


FIG 3

A565-1. SPAN OF ARMATURE COIL

A565-2. CONNECTION OF A GROUP OF COILS

A565-3. INDUCTION MOTOR CONNECTION DIAGRAM

main leads two wires connected to each, it is a delta connection.

Having determined the foregoing, you proceed to strip your motor, taking care not to damage the slots while doing this. If the burnout was caused from the rotor rubbing on the stator laminations, you no doubt will have some of the iron dragged over, which will close the slots at the places where the rubbing occurred. If this is the case, you will have to straighten the slots, and if any rough places occur they can be smoothed down

with a knife blade file, taking care not to short-circuit the iron, which, if short-circuited, will cause heating. Having stripped the motor of all the coils, you must thoroughly clean all the slots and proceed to cut your slot insulation.

The size and amount, you can determine from the old insulation in the slot.

A good practice for 550 v. is to use two pieces of 5-mil: fish paper and one piece of empire cloth of the same size in each slot, placing the empire cloth between the fish papers. In order to make winding easy and speedy, cut your empire and one fish paper just high enough to reach the lower side of the lap on your slot and one fish paper about ¾ or 1 in. higher than the lap on the slots, using this as a guide paper which also protects your coils while sliding them in the slots which would otherwise damage the insulation on the wire if put in next to the bare-iron at the slot opening.

Cut two papers to reach this shoulder only and one to project, placing the two short papers next to the iron in the slot and the guide paper on the inner side of the slot. Now proceed to make a wooden form to wind your coils on; make the form to fit the old coil as nearly as possible, for if the coils are made too long, you will not be able to put your end shields on the motor; if they are made too short, your motor is likely to heat, due to the decrease in the length of your conductor, at the same voltage. It is good practice to tape the lap on the coils with linen tape; some winders only tape the extreme ends of the coils which form a cushion when the coils are placed in position.

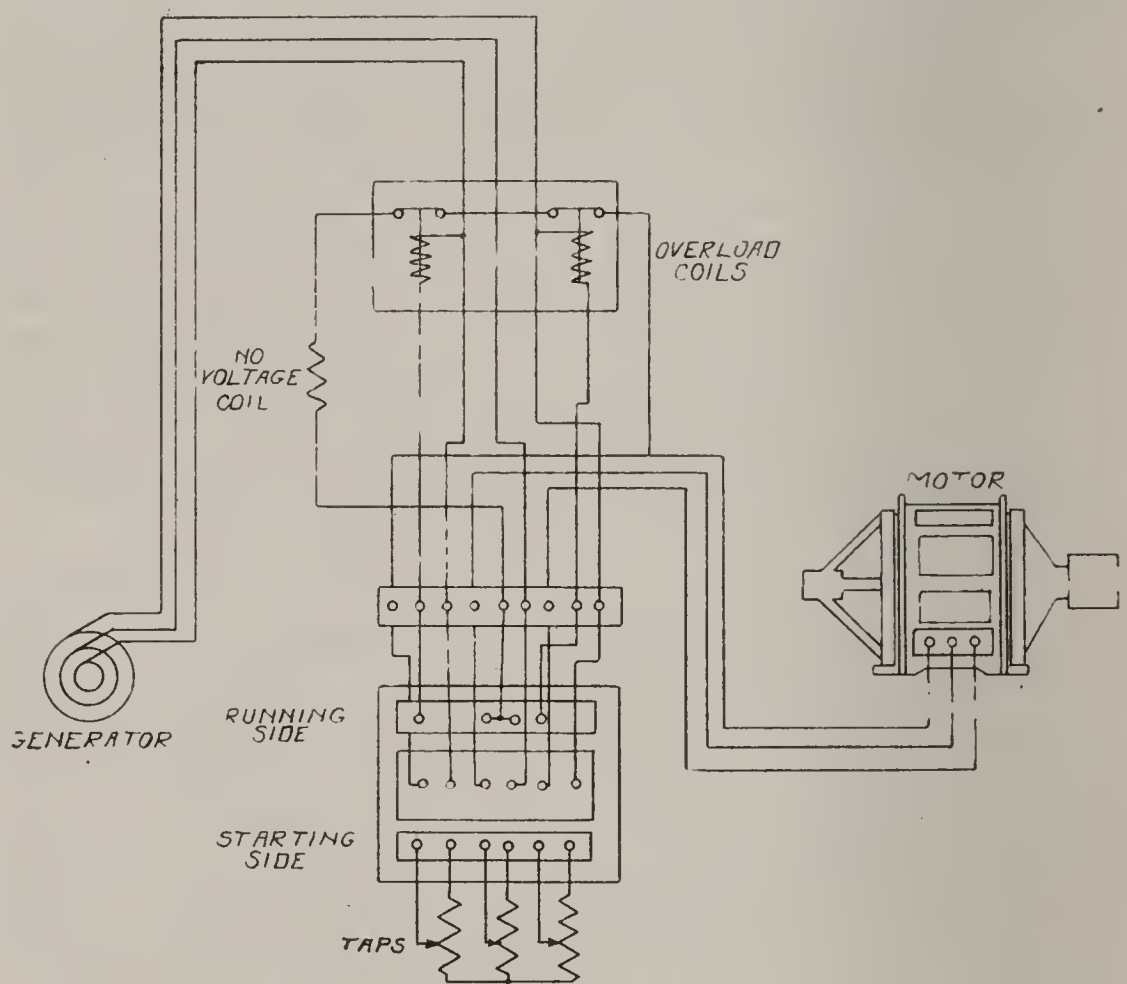
Next place your coils in the motor, and connect each group in straight series which will give you three short connections, also at the end and start of the group you will have one lead, meaning two leads to each group. See Fig. A564-2.

Phase insulation must be placed between groups, to insulate these groups from each other.

When all coils are connected into groups, there will be 24 leads. Proceed to connect each group to its own by skipping two groups and going into the third one, thereby connecting each phase independently, as a 3-phase motor consists of three independent windings. See Fig. A564-3.

566. Q.—Draw diagram of connections of a starting compensator for 3-phase induction motor of a delta connected system, showing overload and low voltage release connections.

A.—Figure A566 shows a diagram of connections from the generator, through the starting box, provided



A566. STARTER CONNECTIONS FOR 3-PHASE DELTA-CONNECTED MOTOR WITH OVERLOAD AND NO-VOLTAGE RELEASE

with no voltage coil and overload coil, of a 3-phase induction motor. The no voltage coil trips the main switch either when voltage fails on the line or when its circuit is opened by either overload coil lifting its plunger and opening the contact of that plunger.

567. Q.—In a 50-hp. Westinghouse alternating-current motor, the rotor will not float from side to side as it should. In fact, the collar on the rotor shaft bears against the box, although it can be pushed over and away from this with little effort.

The motor is level and in line with the driven shaft. I have tried shifting the motor, but without results.

A.—Probably the difficulty with your motor is that the core of the rotor is not quite central with the core of the stator. The tendency is for the rotor core to run central with the stator, in a line along the axis of the motor shaft, and if it is not central, it will try to pull the shaft over so as to run central.

From what you say about the pressure required to throw the motor over, the pull is not very great, and the rotor is not far out; but apparently it is not quite where it should be on the shaft.

The remedy would be, if possible, to shift the rotor on the shaft away from the pulley. This would tend to pull the whole shaft over toward the pulley, and probably would remedy the end thrust.

You seem to have traced out all possible mechanical causes for the endwise thrust, so that we think this magnetic pull must be the trouble.

568. Q.—WHILE STARTING under no load, the starting box of an induction motor bearing the following name-plate data: Make, Ft. Wayne; Type S-4; Form D; R.p.m., 1800; Amp., 49; Volts, 220; Hp. 10; Phase, 1; Cycles, 60, has burned out twice. Split phase starting is used.

On account of being a secondhand machine and receiving but little attention, at present the motor is quite dirty and the stator windings appear to be oil soaked.

After the first burn-out, the rotor was cleaned by a local electrician; this, however, did not eliminate a repetition of the first trouble.

I would greatly appreciate any information you may be able to give me regarding the cause.

A.—The motor in question is an induction type single-phase machine with 3-phase winding, and using a reactance resistance type starting box. As this motor is secondhand and has formerly been overloaded, you may find that some rotor connections are unsoldered, making dead points, and giving the motor a low starting torque. Possibly the operator has held the starting box in starting position too long and the heavy starting current has burned out the insulation on the box; this would also have a tendency to overheat the motor. We would suggest that the stator winding be thoroughly cleaned and the rotor resoldered by using either hard

or silver solder. Also see that the bearings are not worn so that they let the rotor strike the stator. If they are badly worn, they should be replaced with new bearings.

Transmission and Wiring

569. Q.—What is a three wire system and why is it used?

A.—A three wire system is, as its name applies, a system of distribution employing the use of three conductors, one of which is the positive, the other the negative, and the third, usually the middle one, the neutral. In standard practice the voltage between the positive and negative is usually 220, while that between either the positive or negative and the neutral is 110.

Employing these two voltages allows of a flexible system by using the 220-v. for power purposes and the 110-v. for lighting. The chief claim made for this system, however, is the economy in copper, but it is found that when provision is made for any unbalancing of the load there is not very much gained in the economy of copper.

570. Q.—If a transmission line has self-induction, what device could be connected to it to neutralize partly the self induction.

A.—Frequently a transmission line having considerable inductive effect is provided with some sort of a synchronous motor having an over-excited field, which will have the same effect as a condenser in neutralizing the self induction of the system.

571. Q.—When is it advisable to install a plant of a lower or higher frequency than 60 cycles per second? What frequency plant should be installed on a transmission line 25 to 50 miles long for the best results, and why?

A.—A higher frequency than 60 cycles is not at present used to any extent in this country. In the early days, of alternating-current work, a frequency of 132 cycles was used, largely because the machines were made up on nearly the design of the direct current machines. The objection to high frequency is that it increases quite considerably the hysteresis and induction losses in generators, transformers, motors and transmission lines.

Sixty cycles is the standard of practice for short transmissions and for local distributing systems. This is found to give a reasonable amount of hysteresis and induction loss, and, at the same time, does not produce a flicker in incandescent lights which is noticeable.

For very long-distance transmissions, the 25-cycle frequency is used, in order to reduce the induction loss in the transmission lines and the loss in transformers, as such losses become rather large in connection with the large amount of power sent over long-distance lines.

The objection to 25-cycle frequency is, that if current is used at this frequency on incandescent lighting circuits, there is a perceptible flicker of the lights which is unpleasant. Also, where these long-distance lines feed into local systems already established, most of the motors and transformers of the local system are being installed for 60 cycles, and the expense of changing over all this system to 25-cycle apparatus would be very great. It is customary, therefore, to install at the local distributing station a frequency changer—which is merely a rotary converter—for changing 25-cycle current to 60 cycles.

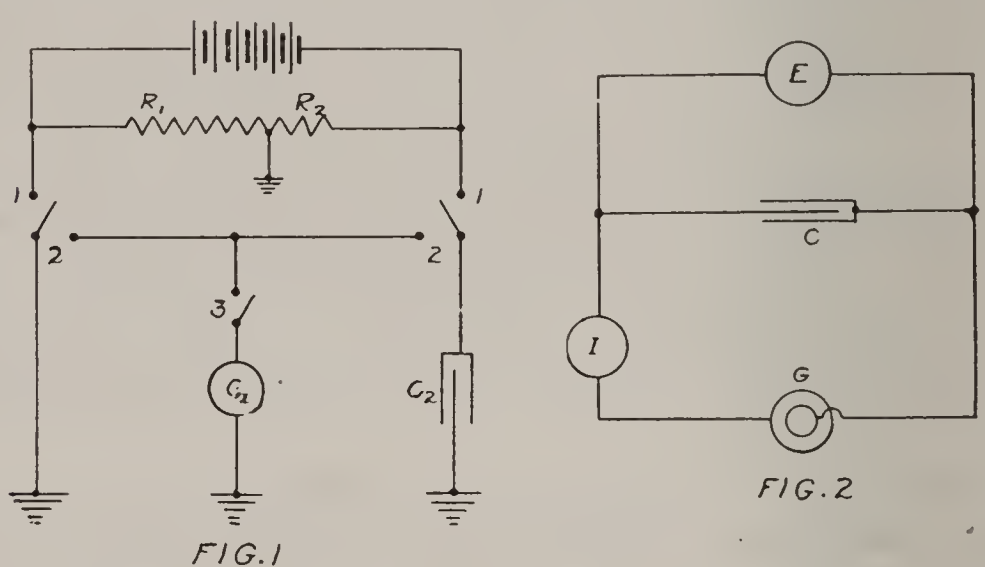
The question of whether to use 25 or 60 cycles becomes merely one of balancing the losses from induction and hysteresis in the transmission line and transformers against the investment and loss in the rotary converter for changing the frequency from 25 to 60 cycles at the distributing station.

For 25 miles—unless the amount of power transmitted is very large—it is doubtful whether it would pay to generate at 25 cycles and then change to 60 cycles. For 50 miles, it is probably about on the dividing line. If the amount of power is considerable, it would pay to generate and transmit at 25 cycles and change to 60 cycles for use.

If there is a composite system, where new apparatus is being installed throughout—a part of the transmission 25 miles and part of it 50 miles—it might be possible to use the 25-cycle transmission throughout and install all apparatus for 25 cycles. There would be some flicker on the incandescent lights, but if people had never had the 60-cycle lighting, this probably would not be sufficient so that there would be any great amount of objection.

572. Q.—Explain how the electrostatic capacity of a cable may be determined.

A.—One of the methods of electrostatic capacity measurements is the Thomson method of mixtures as indicated in Fig. A572-1, herewith. C_1 is a cable, transmission line or other capacity to be measured, and C_2 is a standard condenser. First the switches are closed at 1, 1, and the condensers charged to potentials corresponding to r_1 and r_2 respectively. After complete charge (a cable may require several minutes), the switches are shifted to 2, 2, and the charges equalized. If, then, the switch at 3 is closed, the deflection of the



A572-1. THOMSON TEST FOR CAPACITY
A572-2. IMPEDANCE TEST FOR CAPACITY

galvanometer will be proportional to the difference of the charges. The operation is repeated with various ratios of r_1/r_2 until there is no deflection. Then,

$$C_1 = C_2 r_2 / r_1$$

In the impedance method, the capacity is computed from the reactance as measured with an ammeter and a voltmeter, with alternating current of known frequency, as shown in Fig. A572-2. Then,

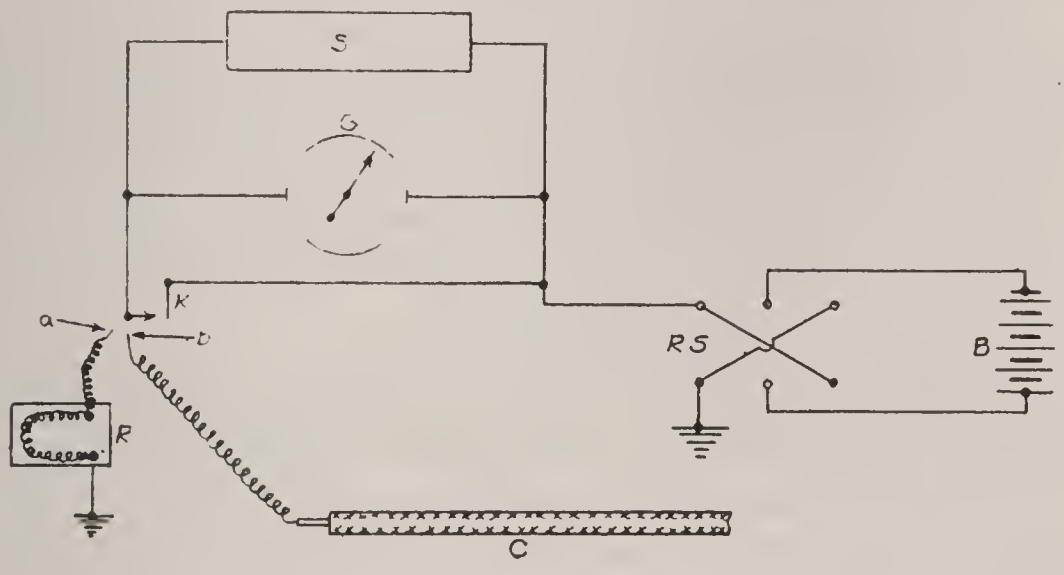
$$C = I\ (10^{-6}) \div (2 \times 3.1416\ f\ E)$$

Where C = capacity in microfarads; E = volts; I = amperes, and f = frequency in cycles per second. Unless the voltmeter is an electrostatic instrument, it should be disconnected when taking current readings. With small capacities, where I is small, care should be taken that the high inductance of a low reading ammeter does not introduce an error. The capacity given by this

method depends upon the wave form. Therefore, the tests should be made from the circuit to which the apparatus being measured will be connected in service.

573. Q.—How may the insulation resistance of a cable be determined?

A.—Apparatus is set up as shown, in Fig. A573, G being a galvanometer with shunt S having connections for 1/10, 1/100, 1/1000, B is a battery of standard testing cells, RS a battery reversing switch, K the galvanometer key, R a standard megohm resistance and C the cable to be tested. If this is installed, the sheath is usually grounded, and the far end of the conductor discon-



A573. TEST FOR INSULATION RESISTANCE

nected during the test, resistance R and switch RS being grounded as shown. If the cable is in coil, it is placed in a tank of water with the ends projecting above the surface and R and RS are connected to the tank, one end of the cable conductor being used for the connection to lead b.

The first connection is made with lead b connected to K, a being disconnected, and S set at its highest value. The shunt is decreased until the largest readable deflection is obtained. A reading is then taken after 1 minute application of battery, and is called d_1.

Next close K, disconnect lead a from it and connect b. Slowly open K and note if there is any reading due to ground current. If this is in the same direction as battery deflection, subtract it from battery reading; if in the opposite direction, add.

Next close K and close RS on one side. After one minute open K and note the galvanometer reading, d_2.

The insulation resistance, R in megohms will equal $d_1 \div d_2$, since R is 1 megohm.

In case it is necessary to use different shunts for d_1 and d_2 to bring the reading on the scale, and if R is not 1 megohm, we should have

$$r = \frac{S_1 \times R \times d_1}{S_2 \times 1{,}000{,}000 \times d_2}$$

Where R is in ohms, and r in megohms.

574. Q.—Explain how the dielectric strength of cable insulation may be found.

A.—The dielectric strength of any insulating wall, cover or path is its ability to resist the passage of a disruptive discharge through it, and is measured by the voltage which must be applied in order to effect such disruptive discharge. High temperature lowers the dielectric strength, and many materials break down at comparatively low temperatures.

Tests for dielectric strength may be made either with a voltmeter or with a spark gap to measure the voltage. In the latter case, the spark gap is adjusted so that it will break down with a certain predetermined voltage and is connected in parallel with the insulation under test. For values of voltage between 10 kv. and 50 kv., a needle spark gap should be used, consisting of new sewing needles supported axially at the ends of linear conductors which are at least twice the length of the gap. There must be a clear space around the gap for a radius of at least twice the length of the gap. Needle gap root-mean-square (R.M.S.) sparking voltages at 25 deg. Cent. and 760 mm. barometer with No. 00 sewing needles are given below:

R.M.S. kilovolts.....	10.0	15.0	20.0	25	30	35	40	45	50
Millimeters gap length.....	11.9	18.4	25.4	33	41	51	62	75	90

575. Q.—How do you find the power factor of a 3-phase alternating-current circuit?

A.—In alternating-current work, the e.m.f. and current are seldom in phase, the current either lags or leads. The power factor, or ratio of volt-amperes to true watts, is the cosine of the angle of lag or lead of

the current. Power factor is the relation of the actual load and the apparent load. Measure voltage, current and watts. The power factor is, watts $\div$ (volts $\times$ amperes).

576. Q.—Why is the power factor of a 220-v. four-wire system lower than that of a 220-v. three-wire system?

A.—Power factor on a system depends more on the load than it does on whether it is a four-wire or a three-wire system. The four-wire system usually carries a two-phase induction motor, which has a much lower power factor than a three-phase motor, such as is used on a three-wire system. This is where the loss in power factor comes in. In a system properly designed, there is no reason why the transmission line for a four-wire system should have a lower power factor than that for the three-wire system. The difference in power factor in the motors comes on account of the difference in the distribution of the wiring on the motors.

577. Q.—The power factor of a 500-v. 3-phase induction motor is 0.85, and at full load the motor takes 25 hp. from the line. What will be the full-load current in each line wire?

A.—In alternating current calculations, we find that the power in watts delivered by any 3-phase balanced circuit is equal to the following formula:

$$P = \sqrt{3}\ E\ I\ p.f.$$

In which P = power in watts;
E = voltage across any 2 lines;
I = current in any main;
p f. = power factor;
$\sqrt{3} = 1.73$.

Transposing the above we have,

$$I = P \div \sqrt{3}\ E\ p.f.$$

which, upon substituting the values given in your problem, becomes,

$$I = (25 \times 746) \div (1.73 \times 500 \times 0.85)$$

or $18{,}650 \div 735.25 = 25.3$ amp. per main.

The above expression (25×746) is used due to the fact that the motor takes 25 hp., and that there are 746 watts in one electrical horsepower, so that the total number of watts is equal to 25×746.

578. Q.—1000KW. IS to be delivered over a distance of 5 mi. to a load consisting of motors and lights. Three-phase, 60-cycle system is used; line loss to be 10 per cent of the power delivered; electromotive force at the end of the line 10,000 v. What size of wire should be used?

A.—In the calculation of this problem, we have neglected regulation, and all capacity or inductance effects. The formula is the same as for question 577.

Power transmitted = 1000 kw.;
Electromotive force at end of line = 10,000 volts;
Distance = 5 miles;
Frequency = 60 cycles;
Loss = 10 per cent, or 100 kw.
Power Factor = unity or 1.

As the total power loss in the circuit is equal to 100 kw., the loss per line wire will be equal to $100 \div 3$ or 33.33 kw. The current per wire is equal to

$$\frac{1000 \times 1000}{\sqrt{3} \times 10{,}000 \times 1} = 57.8 \text{ amp.}$$

The resistance per wire is equal to $\frac{33{,}330}{(57.8)^2} = 9.9$ ohms, which divided by 5, the length of the circuit expressed in miles, is equal to 1.99 ohms per mile. The resistance per mile of a No. 5 B. & S. copper wire is equal to 1.655 ohms. The resistance per mile of a No. 6 B. & S. wire is equal to 2.087 ohms, so that it would be advisable to use a No. 6 B. & S. wire.

579. Q.—WHAT SIZE wire is required to transmit 800 hp. 8000 ft. at 2200 v., 80 per cent power factor and 5 per cent loss? What is the formula and how is it worked out?

A.—You have failed to state whether your system is single, two or three-phase, but we have assumed that it is three-phase, and have based calculations accordingly. In three-phase, alternating-current work, the current flow is calculated by means of the formula of question 577, which gives,

$$I = \frac{\text{kw.} \times 580}{\text{E} \times \text{p.f.}}$$

Where I is the current in amperes; kw. is the kilowatt load; E is the voltage, and p.f. is the power factor.

One electrical horsepower is equal to .746 kw., so that your total load is equal to .746 × 800, or 596.8 kw. We therefore have

$$I = \frac{596.8 \times 580}{2200 \times 0.8} = 196.6 \text{ amp.}$$

Knowing the value of I, or the current flow, the size of the wire may be calculated by the employment of this formula:

$$A = \frac{10.8 \times I \times d \times \sqrt{3}}{e} = \frac{18.7 \times I \times d}{e}$$

In this formula, I is the current in amperes; d is the distance the power is to be transmitted, or 8000 ft.; e is the drop in volts, or in this case 110; A is the area of the conductor in circular mills. Substituting these various values in the above formula we have:

$$A = \frac{18.7 \times 196.6 \times 8000}{110} = 267{,}385 \text{ circular mills}$$

This will require stranded conductor, half the length to be 250,000 c.m. and half 300,000 c.m. to get about the resistance determined by the calculation.

580. Q.—What is the smallest gage wire that can be used in an interior electric lighting system?

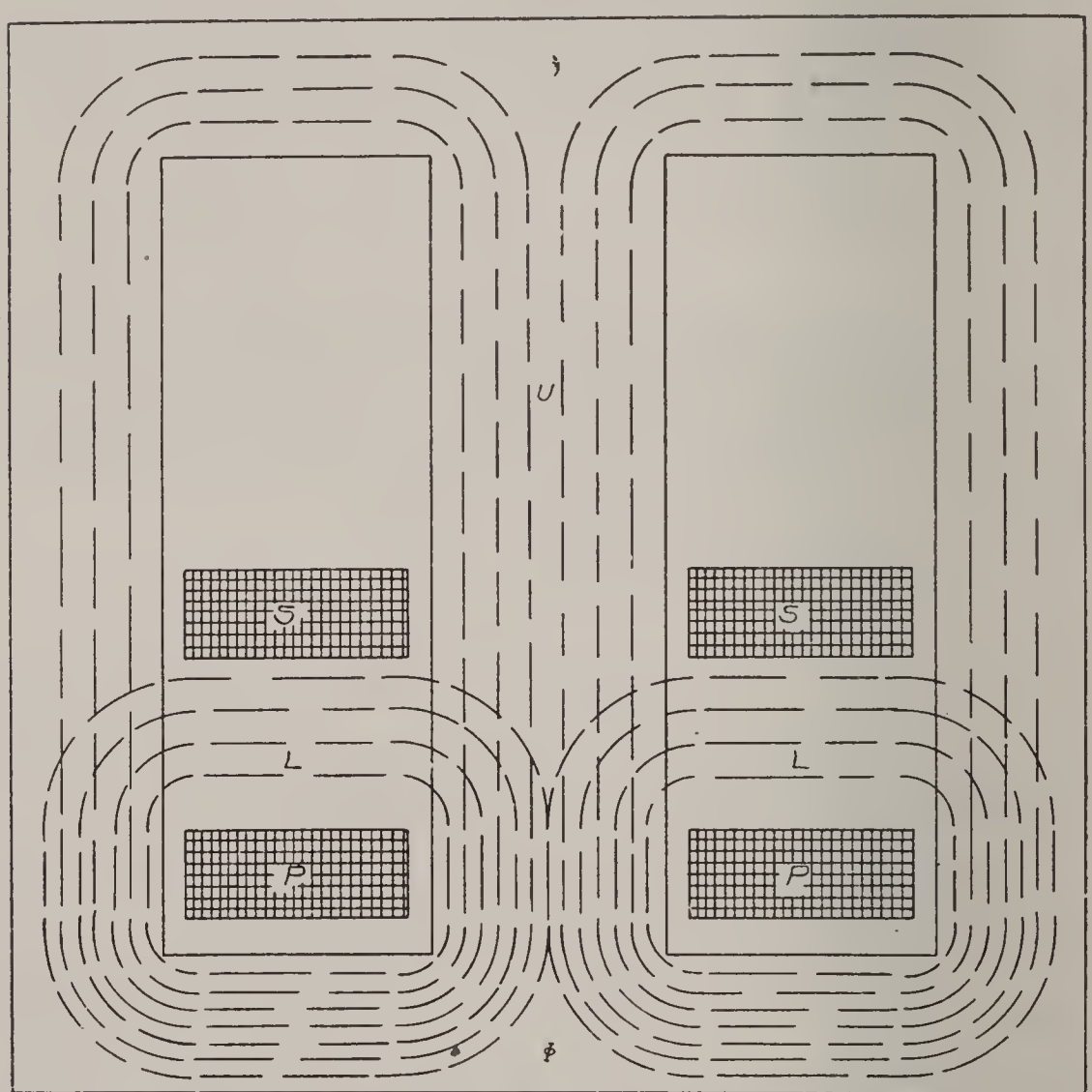
A.—The smallest gage wire permitted for interior wiring is No. 14 B. & S., except for fixture wiring, where No. 16 may be used, or occasionally as small as No. 20. The carrying capacity of this wire is specified by the table of the American Institute of Electrical Engineers, for rubber covered, No. 16, 6 amp.; No. 14, 12 amp. For water-proof, No. 16, 8 amp.; No. 14, 16 amp.

581. Q.—What is the maximum load in watts that should be put on one interior lighting circuit?

A.—Not over 660 watts should be carried on one cut-out, and not over 12 standard size sockets on any one circuit.

582. Q.—What is the theory of the action of the constant current transformer?

A.—The action of the transformer may be explained with the aid of Fig. A582, as follows: Primary coil P and secondary coil S of the transformer surround a long laminated iron core. This core, and the yokes at the top, bottom and sides form a double magnetic circuit. The magnetic flux *F* which at a given instant passes through the primary coil flows partly through the



A582. SECTION OF CONSTANT-CURRENT TRANSFORMER

secondary coil as the useful flux U, and partly leaks across between the primary and secondary coils as the leakage flux L.

Leakage flux L, in flowing across the air spaces from core to yokes, constitutes an intense magnetic field which pushes up on the secondary coil. The secondary coil is suspended and partly counterbalanced by a weight, so that the upward push of the leakage flux just suffices to sustain the coil. If the resistance of the secondary circuit is increased, the immediate result is to reduce the secondary current below its normal

value, which lessens the upward push of the leakage flux on the secondary coil. The secondary coil then, owing to the unbalanced action of the weight, moves down towards P; the leakage flux is lessened in amount and the useful flux U is increased in amount. This increase of useful flux increases the induced electromotive force in S; and the downward movement of S continues until the induced electromotive force in S is large enough to produce the normal value of the current through the increased secondary resistance.

Similarly, a decrease of resistance of the secondary receiving circuit causes a momentary increase of secondary current which increases the upward push on the secondary coil. This coil moves upward until the secondary current is reduced to the normal value.

583. Q.—Is a constant current (tub) transformer started with the secondary short circuited? If not, how is it started?

A.—Whether or not the secondary is short-circuited depends upon the design of the transformer and the character of the load to which it is connected. Various transformer manufacturers have views which differ on this subject.

In one system for constant current control of a series incandescent street lighting circuit, a series alternating regulator, consisting of an E-shaped core and one coil, is employed in connection with the potential transformer, and starting is accomplished with the load short-circuited. When used with arc lamps, this regulator is provided with a starting switch and reactance so arranged that the regulator is brought to a position of maximum reactance with the arc lamp line short-circuited. This short-circuit being opened, the regulator moves to its normal position without allowing the lamps to chatter. With incandescent lamps in the line, the starting arrangement is similar, but a device is used which allows for the lower resistance which the lamps have when cold than when carrying their normal current.

In this system, the reactance coil, placed on the rear of the switchboard and connected in series, reduces the initial flow of current through the regulator, which is direct-connected to the transformer, until the regulator has automatically introduced impedance of its own into the circuit.

Other systems are designed to start with full load on the secondary, but provision is usually made so that short-circuiting of the secondary will not cause any serious damage.

584. Q.—When several circuits of series lamps are connected to a transformer, and one circuit is to be cut out, is the switch at the transformer opened or short-circuited? Why?

A.—To cut out one circuit, a single-pole switch is closed, thus short-circuiting the line that carries that part of the load. Before cutting out this circuit, however, some arrangement should be made, automatically or otherwise, to increase the reactance in the circuit. One system employs a pedal attachment, which, by a system of levers, forces the core to occupy a position farther within the coil and adds sufficient reactance to prevent a momentary heavy rush of current. After the switching operation is completed, the pedal is released.

585. Q.—What is the maximum voltage you would expect to find on a constant-current transformer secondary?

A.—The approximate maximum voltage expected to be found on a constant current transformer secondary may range from 110 to 3000 v., depending upon the number and type of lamps connected in the circuit.

586. Q.—I have at present installed a 5 kw. single-phase transformer, which is loaded to its full capacity with lights. I wish to install a 1-hp. 3-phase motor near this same point. The question is, can I put a 1-kw. transformer beside the 5-kw. connecting in open delta and operate my three phases satisfactorily for the above motor, without detriment to the service on the large transformer or injury to the small one?

A.—The connection which you intend to make between the 5 kv.a. and the 1 kv.a. transformer is one which is used to considerable extent, but when so used, the two transformers are generally of the same capacity. By using two transformers of such great difference in capacity, you are introducing difficulties which will be hard to overcome.

The total load which you now have on the 5 kw. transformer is 5000 watts, or full load. At unity power

factor a one horsepower motor will require 746 watts, making a total of 5746 watts to be placed on the two transformers.

Assuming that these transformers are the same size, then the capacity of each will have to be 5746 ÷ 2, or 2873 w.

To supply the necessary kv.a., due to the lagging current with this type of connection, at least 16 per cent should be added to the above rating, or a total of 3352 v.a. should be provided for each transformer, making a combined volt-ampere rating of 6704, which you will see is somewhat greater than the combined capacities of the two transformers which you have at the present time.

If, however, the motor is not continually operating under full load, and the lighting load does not, in any way, conflict with the motor load, you may be able to employ your scheme. Under those conditions, connect the two transformers in open delta, with the motor connected in the regular way. Having done this, and the lighting load entirely disconnected, determine the v.a. load on each transformer by means of indicating ammeters and voltmeters. Then gradually connect the lighting load, distributing it as much as possible among the three phases, so that by observing your instruments and the v.a. load on each transformer, you will readily be able to tell when the small transformer is carrying its full load. The remaining load can then be connected on to the 5 kv.a. transformer.

While we stated above that this scheme can be employed, we would hardly advise you to use it. If you have another 5 kv.a. transformer handy, we would strongly advise you to connect that in open delta with the present 5 kv.a. transformer.

587. Q.—If a ground occurs on the secondary side of two or more transformers fed by one primary line (these transformers are not banked but feed separate lines which supply residences and stores in different parts of the town), will this show up on a ground detector installed and connected to one of the transformers, or will it only show if all the above transformers are banked together?

A.—If a ground occurs on the secondary of one of two transformers which have the primaries in parallel,

this ground will not show on a test of the primary circuit, nor would it show on a test of the secondary of the other transformer. The test for a ground is made by completing a circuit from the other side of the distribution system to the ground, which would give a complete circuit from the source of current. In the case that you have outlined, there would be no return circuit through the ground from the primary circuit or from the secondary of the ungrounded transformer.

588. Q.—How OFTEN should the oil in service transformers be changed?

Does dirty oil have a tendency to lower the efficiency of the transformer?

A.—Presence of dirt in transformer oil will have a tendency to reduce the efficiency of the transformer, first by interference with the proper circulation of the oil, and cooling, and second, by the deposit of such dirt upon the outside of the coils, both of which would tend to reduce the proper dissipation of the heat.

Of course, it would be only when an excessive amount of dirt is present, that the efficiency would be reduced materially, but insulating value is greatly diminished by even a small amount of dirt.

The frequency of change of transformer oil depends entirely upon local conditions, and it requires the use of good judgment as to when the oil is of no further use.

One of the worst things to contend with in the operation of oil-cooled transformers is the presence of water in the oil. High voltage transformers should be tested at least once a month for this condition, and if the oil is below par, it should be dehydrated. The best method for doing this is by the use of a filter press dehydrater. In this type, the oil is pumped through several thicknesses of blotting paper. When using this apparatus, it should be seen that the paper is carefully dried, and it is better to soak it, first of all, in clean oil that is perfectly dry.

When the oil is in very bad condition the paper should be changed, from time to time, as often as found necessary.

589. Q.—A TRANSFORMER is supplied with a primary electromotive force of 1000 v. and has 1000 turns on the primary. How many turns must there be in the

secondary to have an electromotive force at the terminals of the secondary of 104 v.?

A.—The values of the primary and secondary voltages of a transformer are directly proportional to the number of turns on their respective coils, or by letting

N_1 = the number of turns on the primary coil;
N_2 = the number of turns on the secondary coil;
E_1 = the primary voltage;
E_2 = the secondary voltage.

We have $N_1 : N_2 :: E_1 : E_2$. In this case, $N_1 = 1000$; $N_2 = ?$; $E_1 = 1000$; $E_2 = 104$; so that by substituting these values in the above, we have $1000 : N_2 :: 1000 : 104$; or $N_2 = 1000 \times 104 \div 1000 = 104$ turns.

590. Q.—A COIL of wire wound on an iron ring has a resistance of 5 ohms, an inductance of 0.5 henry and an alternating electromotive force of 110 v. at 30 cycles per second is applied. What current will flow through the coil?

A.—The value of the alternating current flowing through a circuit containing inductance may be represented by the following formula:

$$I = E \div \sqrt{R^2 + (2 \times 3.1416\ f\ L)^2}$$

Where I = current in amperes;
E = voltage;
R = resistance of circuit, in ohms;
f = frequency, in cycles per second;
L = inductance, in henrys.

Substituting the values given, we have

$$I = 110 \div \sqrt{5^2 + (2 \times 3.1416 \times 30 \times 0.5)^2}$$
$$= 110 \div \sqrt{5^2 + 94.24^2} = 110 \div 94.4 = 1.06 \text{ amp.}$$

591. Q.—A CURRENT of 10 amp. flows 1 hr. through a resistance of 50 ohms. What is the rate in horsepower at which work is done?

A.—By Ohms law the voltage is equal to the current in amperes times the resistance in ohms, or $E = I\ R$, or $50 \times 10 = 500$ volts.

Watts power is equal to volts times amperes, or $500 \times 10 = 5000$ watts. 746 watts = 1 horsepower. Then $5000 \div 746 = 6.8$ hp. = rate in horsepower at which work is done.

592. Q.—A GROUP of three circuits in multiple is connected across the lines of a 100-v. circuit. The first

circuit has a resistance of 10 ohms, the second 25 ohms, and the third of 50 ohms. What is the joint resistance of the three paths in multiple? What is the current in each circuit? What is the rate in watts at which work is being done in each of the circuits?

A.—The resistance of a multiple circuit is found as follows: Divide one by the resistance of each individual member; add these fractions; the reciprocal of the sum will be the total resistance.

In your case we have, $1/10 + 1/25 + 1/50$, which equals 8/50. The reciprocal will be 50/8 or 6.25 ohms.

Current in each member of the circuit will be found by dividing the voltage by the resistance. For the first circuit, $100 \div 10$ gives 10 amp.; for the second, $100 \div 25$ gives 4 amp.; for the third, $100 \div 50$ gives 2 amp.

The watts are found by multiplying the current by the voltage. For No. 1 circuit, $10 \times 100 = 1000$ watts; for No. 2 circuit, $4 \times 100 = 400$ watts; for No. 3 circuit, $2 \times 100 = 200$ watts.

593. Q.—I WISH to heat a room 7 by 9 ft. and it gets as cold as 18 deg. below zero in winter; voltage 220, amps. 5, what size wire and how long would it have to be? May any kind of wire be used?

A.—The amount of heat required for your room, assuming that it is 8 ft. high, and that ⅓ of the area, approximately, is glass, would be 24,800 B.t.u. per hour. Five amperes, at 220 volts, will give you the equivalent of 3756 B.t.u. per hour, which, as you will see, is only little more than 1/7 of that needed to heat the room in 18-deg.-below-zero weather. In other words, it would be necessary to use about 40 amp., instead of 5 amp., through your resistance to give you the required heat.

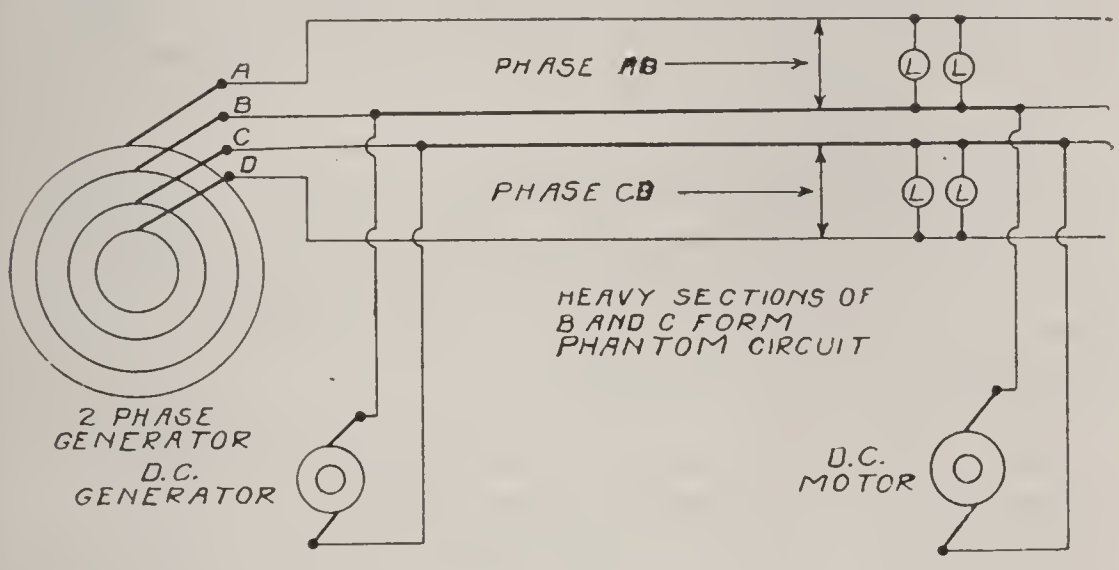
The resistance required for iron wire to carry 5 amp. at 220 volts would be 44 ohms. Number 15 iron wire, mounted in a wood frame, so that air can circulate around it to carry away the heat, will carry 5 amp., and it will take about 2000 ft. of this wire to make 44 ohms.

To give you 40 amp. at 220 volts, you would need about 5.5 ohms. To carry this current safely, you will need two No. 8 wires wound in parallel, and each wire will have to have 11 ohms resistance. For No. 8 wire, it takes 250 ft. per ohm, or 2750 ft. for 11 ohms, so that

to get your 40 amp., you would need 5500 ft. of No. 8 iron wire wound two in parallel.

594. Q.—What is a "phantom" circuit?

A.—In order to give a clear understanding of this, Fig. A594 shows a 2-phase alternating-current generator connected to its load, consisting of a number of lamps marked L. This generator is represented by the four concentric rings connected to the brushes A, B, C and D. By connecting this generator to the load through the lines shown, we have two phases, one known as AB, and the other as CD.



A594. DIAGRAM OF PHANTOM CIRCUIT

Now, if the windings of this generator are not interconnected, it will be found that there will be no potential difference existing between lines A and C or D. Also, no potential difference between B and C or D, and, in a like manner, it will be impossible to find any potential difference existing between D and B or A, also C, and B or A. In other words, AB and CD are two entirely separate circuits.

That being the case, it is possible to connect a direct-current generator to lines B and C, and transmit through them current to a direct-current motor, as shown on the left of the sketch. This would in no way interfere with the operation of the alternating-current circuits, nor with the operation of the direct current supply. This same thing could just as readily be done by connecting a direct-current generator and a direct-current motor to lines A and D, instead of B and C.

While the above can very easily be used, it is not to any extent employed in practice. The greatest field of the "phantom" circuit is found in telephone work. Telephone circuits already in existence may be so combined as to produce additional circuits without using any additional wire. As in the case just cited, the principle of the phantom circuit in telephone work rests in the fact that it is possible to use the limbs of two metallic circuits for a third circuit.

595. Q.—An electrolytic lighting arrester is normally charged to a potential about equal to that of the line which it serves. In case it becomes necessary to open such an arrester for inspection or repairs, would it first be necessary to discharge it? If so, how is this done?

A.—An electrolytic lightning arrester does not retain any of the voltage which is passed through it and therefore does not need to be discharged before working on it. The voltage is passed through it to maintain a proper film on the electrolyte and as the arrester is properly grounded no static is retained.

596. Q.—Will direct or alternating current cause corrosion of underground water pipes?

A.—Tests made to determine whether alternating currents have tendency toward electrolytic action in water pipe show that they do. The extent of such action is, however, less than 1 per cent of that which would be produced by a corresponding direct current.

597. Q.—What is the difference between alternating and direct current?

A.—A direct, or continuous current is one, as its name implies, which travels in one direction, while in the case of an alternating current, the electromotive force and the current are reversed at stated intervals—that is, the current will flow first in one direction, and then will be reversed and flow in the opposite direction. These reversals, in commercial practice, take place at a frequency of about 120 times per second for distributing systems, and 50 times per second for long transmission lines.

Instruments

598. Q.—Is a special voltmeter necessary for each kind of alternating-current circuit?

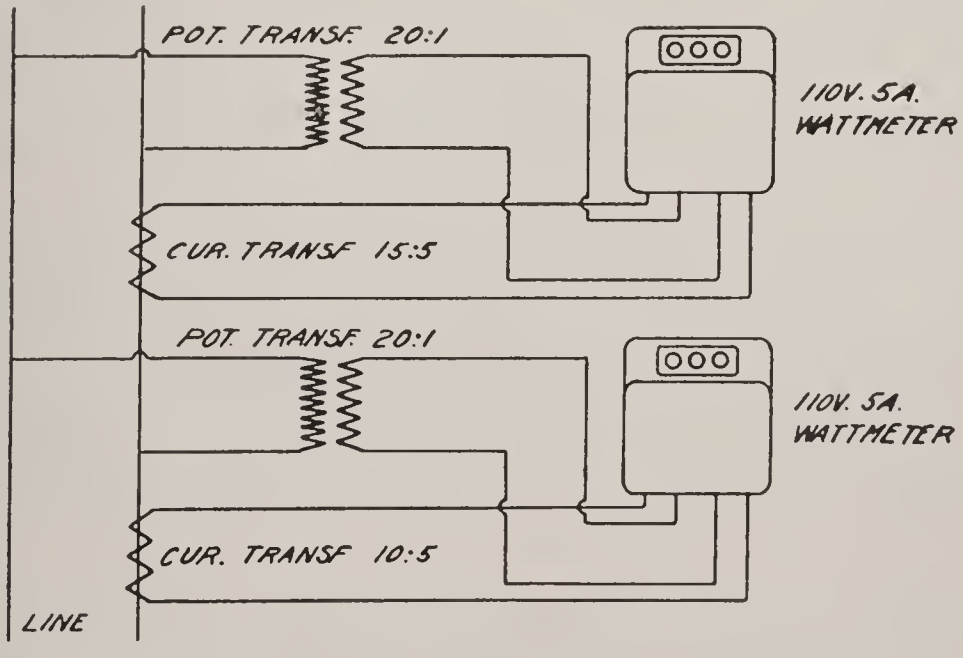
A.—The same voltmeter can be used for single-phase, 2-phase or 3-phase lines, but a direct-current voltmeter will not register correctly on a.c. lines.

599. Q.—Please give the formula or constant for finding the total kilowatt output of a 3-phase generator by timing the disk rotations of a 2300-v. Westinghouse polyphase integrating wattmeter; also the method of finding the disk constant or various size meters.

A.—All Westinghouse meters run 25 r.p.m. with full load passing through them. Full load in the case of a polyphase meter would mean twice the product of the current and voltage capacities marked on the meter. For the last 2 or 3 yr. these meters have had a small paster on the inside of the terminal cover which gave the kilowatt-hours per revolution of the main disk.

You can obtain the kilowatt-hours per revolution from the above data or from the marking on the paster in case the meter was originally supplied with this information. Knowing the kilowatt-hours per revolution, you can readily determine the kilowatts flowing at any given time by counting the revolutions for a short period, multiplying by the kw.-hr. per revolution and dividing by the time expressed as hours.

600. Q.—By what constant must we multiply the meter readings to get the correct energy output with the connections shown in Fig. A600? How can we get these constants, or how are they figured out on this or any other meter?



A600. Connections of Wattmeters

How do you figure the constant on a meter that is taken off from 2300-v. 60 to 5-amp. circuit and put on to a 110-v. 5-amp. house circuit?

A.—In the diagram of connections sent us, we note that you have two watt-hour meters connected to a single phase line. Both of these meters, as you state, are common 110-v. 5-amp. house meters—one of which is connected to the line through a 20 to 1 potential transformer and 15 to 5 current transformer, while the other is connected to the line through a 20 to 1 potential transformer and a 10 to 5 current transformer.

These meters, as you state, are 110-v., 5-amp. instruments, and as a consequence, if they were connected to such a 110-v. line, they would indicate directly the amount of energy being measured. There is, however, a transformation of 20 to 1 in the case of your voltage, and another transformation of 15 to 5, or 3 to 1, in the case of your current; so that for every kilowatt indicated by your wattmeter, there are 20 times 3 or 60 kw. passed over your system; so that it will be necessary to multiply all your meter readings, in the case of this meter, by 60 to obtain the true number of watts or kilowatts, as the case may be.

With the other meter, however, we note that you are using a current transformer having a ratio of 10 to 5, or 2 to 1. This would require, instead of a constant of 60, a constant of 20 times 2 or 40, thus necessitating multiplying the readings of this instrument by 40.

Undoubtedly the meter referred to in your second question was primarily designed and constructed for use on a 2300-v. 60-amp. circuit. If such is the case, the meter, while not actually measuring the energy passing through the circuit, at least indicates the actual amount of energy passing through the circuit.

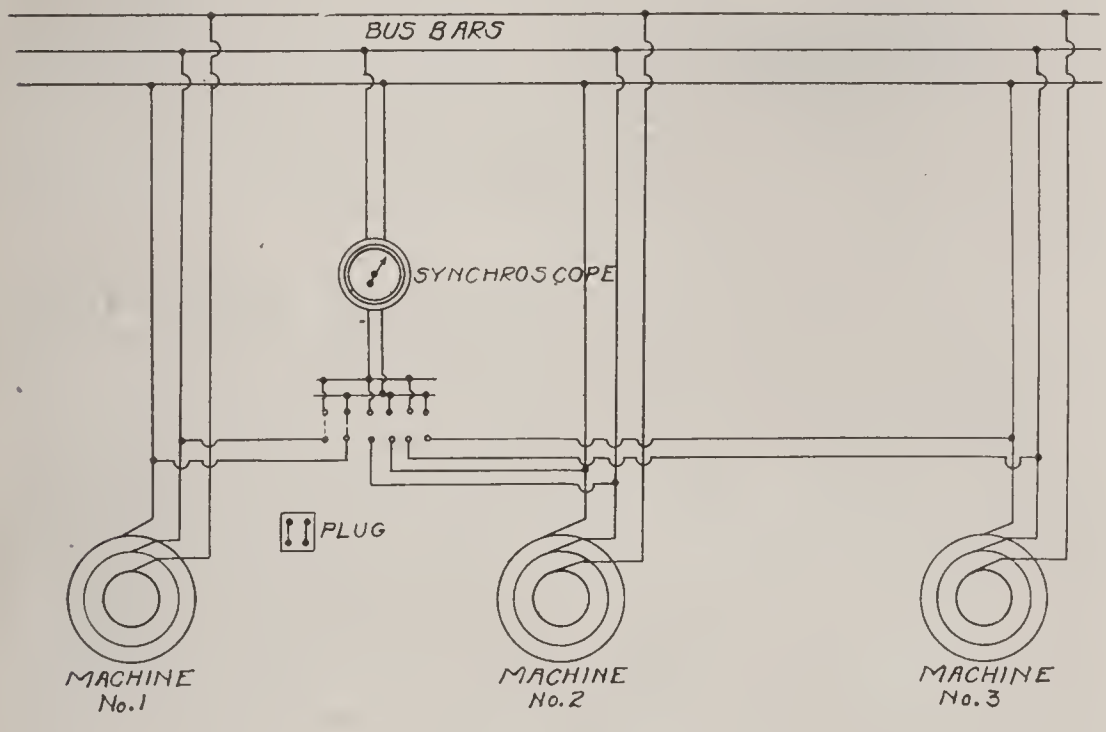
If, therefore, you intend to use this meter on a 110-v., 5-amp. circuit, it will be necessary to make corrections accordingly. The voltage transformation in this case is 1 to 20, while your current transformation is 1 to 12. It will, therefore, be necessary, in order to obtain the correct reading, to multiply your wattmeter reading by 1/240.

601. Q.—Reading of a recording wattmeter is 8900 watt-hours at the beginning of the month and 25,000 watt-hours at the end. The meter constant is 2. With

power at 5 cents per horsepower-hour, what should be the bill for the month?

A.—The total number of watt-hours of energy consumed, between the two meter readings is 25,000 — 8900 = 16,100 watt-hours. The constant of this meter is, however, 2, so that the actual energy consumed will be equal to 2 × 16,100 or 32,200 watt-hours. One horsepower-hour is equal to 746 watt-hours, so that the total number of electrical horsepower-hours delivered will be equal to 32,200 ÷ 746 or 43.1 hp. hr., which at the rate of 5 cents per horsepower-hour will result in a bill of $2.16.

602. Q.—Please give diagram for connections of three 3-phase a.c. generators, and a synchroscope.



A602. CONNECTIONS OF GENERATORS AND SYNCHROSCOPE

A.—The diagram of connections in Fig. A602 will show the proper method of connection for a synchronoscope for use with three 3-phase machines.

One side of the instrument is connected directly across the bus bars, only one phase being necessary, due to the fixed relation between this and the other two phases. The other side of the instrument is connected to a synchronizing bus which in turn is connected to the synchronizing plug receptacles as shown; the same phase of each generator is then connected to corresponding synchronizing plug receptacles so that with the plug

inserted for synchronizing machine No. 1 the circuit would be completed as shown by the dotted lines.

603. Q.—We have two Allis-Chalmers turbo-generators, one of 140 kv.a. and the other of 240 kv.a. rating, both of which, however, operate at 3600 r.p.m. and deliver current at 440 v., 60 cycles.

When paralleling these machines considerable disturbance occurs between the two machines and when the synchronizing plug is withdrawn the indicator needle recedes from the "slow" and "fast" marks. The indicator is provided with one running and one starting plug and is of General Electric make.

A.—When alternating-current generators are in synchronism, a synchronism indicator connected thereto will, or should if correct, indicate that the machines are in synchronism. If, however, the plug is removed, the instrument is no longer under the influence of these machines, or, in other words, it is "dead," and as a result, due undoubtedly to an unbalanced condition of the revolving member, the indicating pointer will drop either in one direction or the other.

If your synchronism indicator is in proper adjustment and all connections are correct, we fail to see why you should experience any trouble when throwing your incoming machine onto the bus bars. The writer has operated a great many alternating-current generators, and has found from his experience that the best results may be obtained by so adjusting the speed of the incoming machine as to cause the synchronism indicator to revolve at a speed of about 2 or 3 r.p.m., and then throw in the switches by so timing them that they close the instant synchronism is indicated. If you allow the instrument to operate faster than 2 to 3 r.p.m., and you start to operate the switch at the instant synchronism is complete, due to the time element bound to be involved, you are unable to obtain quick enough action to cause the switch to come in on time. As a consequence, you will experience the trouble you mention.

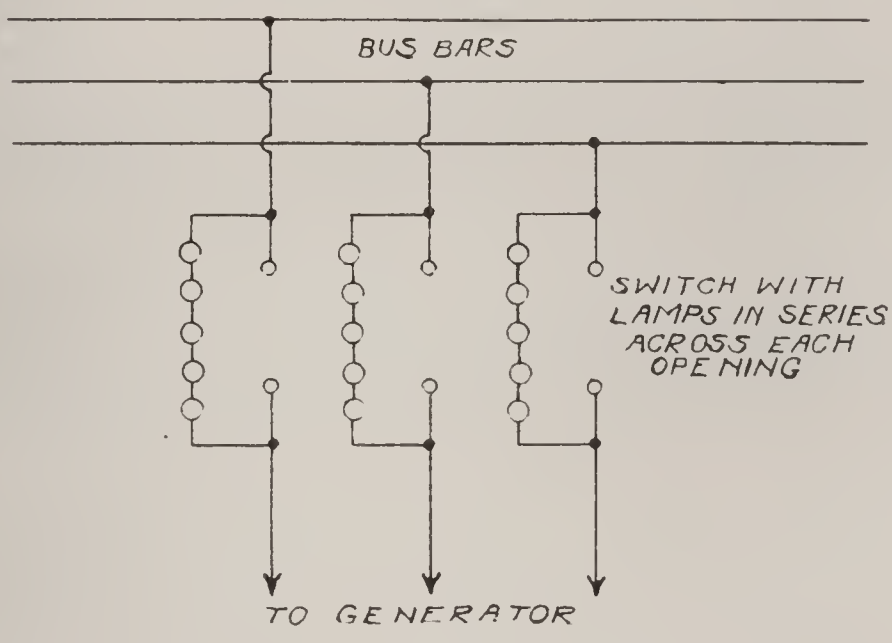
604. Q.—What are the proper connections for a bank of lamps for synchronizing three 600-v., 3-phase generators?

A.—The proper connections for synchronizing lamps, depends on two things—(a) whether your generators

have been properly "phased in" and (b) whether you wish to synchronize "light" or "dark."

As "dark" synchronizing seems to be the most popular system, Fig. A604 shows diagrammatically the proper connection for that method.

As the machine nears the proper speed, the lamps will flicker more and more slowly—the switch should be closed when lamps are dark.



A604. CONNECTIONS FOR SYNCHRONIZING LAMPS

In case the three sets of lamps do not light up and get dark simultaneously, your generator phases are wrong—and this is remedied by reversing connections of any one pair of your generator leads.

After your machine has once been connected properly, i. e., so that the lamps flicker simultaneously, two sets of lamps can be dispensed with, as the remaining one set will indicate the relations of one phase. If one phase is in synchronism, the other two must be, provided, of course, that the machine has previously been "phased out" with the three sets of lamps as above.

The number of generators makes no difference. The first machine is, of course, connected to the bus bars without synchronizing, after which each of the succeeding ones are in turn paralleled with the bus bars as above.

605. Q.—What is the rule for figuring the pull on a solenoid? If a thin brass tube 2 in. diameter and 6 ft. long is wound with 1400 turns of wire, what pull will

be exerted by a wrought-iron core inside it with a current of 5 amp. flowing in the winding?

A.—The formula for figuring the pull of a solenoid is as follows:

$$P = C\ A\ I\ N \div L$$

in which P = pull in pounds
C = a constant
A = area of core, sq. in.
I = current, amp.
N = number of turns
L = length in inches.

From curves we find that the constant for your particular problem is 0.03; then substituting figures in the formula, we obtain the following result:

$$P = 0.03 \times 3.14 \times 5 \times 1400 \div 72 = 9.02 \text{ lb.}$$

The proportions of your magnet are not in accordance with good engineering practice, and if you should decrease the length to about 1 ft., using the same number of turns and current, you will get nearly six times the pull.

PART IV

Refrigeration

606. Q.—What is the principle of refrigeration?

A.—The fundamental principle of refrigeration is the abstraction of heat from the body, or bodies, to be cooled. Refrigerating plants may be divided into four classes:

In the machines of the first class, a permanent gas is used as the working fluid. After being compressed and cooled, this working fluid is expanded adiabatically and its temperature reduced to a low value.

Machines of the second class, called "vapor compression" machines, liquefy a vapor by the application of pressure, and by the subsequent re-evaporation of this liquid under low pressure, the desired low temperature is obtained.

Refrigeration plants of the third class are called "absorption" plants. In such plants, a volatile vapor, like ammonia, is absorbed by water or other liquid, and then driven off under high pressure by heat, in such a manner that it may be afterwards cooled and condensed. It is then evaporated under low pressure, producing a low temperature.

In apparatus of the fourth class, the gas is compressed to a very high pressure and cooled. When it is subsequently expanded, the work done in separating the particles against their mutual attraction lowers the temperature.

607. Q.—What is the boiling point of ammonia?

A.—The boiling point of ammonia depends on the pressure. We presume you refer to the boiling point at atmospheric pressure. This would be —27 deg. F.

608. Q.—What is a safe tempeerature at which to store ammonia?

A.—It is necessary to store ammonia always under pressure, and at ordinary atmospheric temperature of about 70 deg., the pressure is 114 lb. gage, so that with a reasonably strong container, there will be no difficulty in storing at this temperature.

609. Q.—What is the difference between ice melting capacity and ice making capacity?

A.—Ice melting capacity is the heat absorbing power of melting ice. The latent heat if ice is 143.5 B.t.u. By this we mean that to melt one pound of ice the heat required is equal to 143.5 B.t.u. To melt one ton, 2000 times 143.5, or 287,000 B.t.u., are required. This is equal to one ton refrigerating capacity. An ammonia compressor which is capable of pumping the vapor given up by the liquid ammonia which has absorbed 287,000 B.t.u. in 24 hr. is called a one-ton refrigerating machine. Then a one-ton refrigerating machine is one which is capable of doing an amount of heat absorbing duty equal to that performed by the melting of one ton of ice from and at 32 deg. F. In actual practice, when ice is used to cool produce in a room or box, the heat absorbed will amount to something more than 287,000 B.t.u., according to temperature of the room. In a room or box held at a temperature of 45 deg. F., the water itself will absorb heat until it reaches the temperature of the room. In this case, each pound of water will take up 13 B.t.u. after the ice is melted. Then the actual refrigerating capacity of a ton of ice melting in this room will equal 313,000 B.t.u.

The ice making capacity of a machine is the capacity it has for handling the ammonia necessary to evaporate during the process of producing ice from the available water and lowering the temperature of the ice to the temperature of the brine in the ice tank. The ice making capacity of a machine will vary according to the temperature of the water used for filling the ice cans in different plants and locations. The water from which the ice is made cannot be colder than the coldest available cooling water unless the machine is used to lower this temperature. So all heat taken from the water (below the temperature of the cooling water) is absorbed by the ammonia which has to be handled by the machine.

Take, for example, a plant where water at 60 deg. F. is used. The temperature of the water is first lowered to 32 deg. F., requiring the withdrawal of 60 minus 32 or 28 B.t.u. per lb. of water. To convert a pound of water at 32 deg. F. into ice at 32 deg. F., calls for the removal of the latent heat, which is 143.5 B.t.u. So, not counting any loss of any kind, it takes 143.5 plus 28

or 171.5 B.t.u. to make a pound of ice at 32 deg. F. from a pound of water at 60 deg. F.

To have ice making profitable, the ice must be made quicker than it is possible to make it at 32 deg. F., so the brine temperature carried is about 14 deg. F. To lower the temperature of one pound of ice from 32 deg. F. to 14 deg. takes $(32 - 14) \times 0.5$, or 9 B.t.u. (specific heat of ice is 0.5), therefore to make one pound of ice at 14 deg. from water at 60 deg. 180.5 B.t.u. must be absorbed. To produce one ton will require 180.5×2000, or 361,000 B.t.u. But in ice making there is a constant loss due to heat leakage through the ice tank insulation, open lids, uncovered pipe work and inserting of cans which have the temperature of the atmosphere of the tank room. All such losses will amount to from 10 to 30 per cent, depending on the condition and construction of the plant. If we take 15 per cent as the general average in a fair plant, it will add a little over 27 B.t.u. to each pound of ice or will make a total of 415,150 B.t.u. taken up by the ammonia in the making of 1 ton. Under these conditions the ice making capacity is about 55 per cent of the refrigerating or ice melting capacity. It is customary to consider the ice making capacity of a plant as half its refrigerating capacity.

610. Q.—What is the B.t.u. equivalent of one ton of refrigeration? Would this be a fair basis on which to sell cold storage?

A.—A ton of refrigeration represents the heat taken up by the melting of 2000 lb. of ice. One pound takes up 143.5 B.t.u. in melting, so that for the ton we shall have to take up 287,000 B.t.u. The standard recently adopted is 288,000 B.t.u. per 24 hr.

Refrigerating machinery is now rated on that standard.

Selling cold storage service on the basis of B.t.u. taken up by the brine would certainly be an entirely equitable method of charging. The price that should be made depends, of course, on the cost of producing refrigeration in your plant under the fuel and labor costs of your operation.

611. Q.—How are refrigerating machines and compressors used for ice making rated?

A.—It is customary to rate refrigeating machines in tons of refigerating capacity in 24-hr. (288,000 B.t.u.) on the basis of a suction pressure of 19-lb. gage, corresponding to 5 deg. F. temperature of saturated ammonia vapor and a condensing pressure of 153-lb gage, corresponding to 86 deg. F. The actual capacity increases with the increase of the suction pressure, and decreases with the increase of the condensing pressure.

Compressors used for ice-making purposes must be of from 90 to 100 per cent greater capacity than those used for refrigeration, due to the fact that, in ice-making, the water must be cooled from the temperature at which it leaves the cooling coil, say 85 deg. F., to the temperature of the freezing tank, which may be 14 deg. F., and in addition to this extract 143.5 B.t.u. of latent heat for each pound of ice.

Exact size of compression cylinders for any given refrigerating capacity cannot be given unless the temperature to be maintained in the cooler be known. For general purposes it is safe to say that a compressor having 7500 cu. in. capacity per minute will handle sufficient gas for one ton of refrigeration during a continuous run of 24 hr. This rule is based on an expansion pressure of 15-lb. gage, with cooling water at 70 deg. F. If the pressure in the expansion coils varies from 15 lb. the number of cubic inches of cylinder capacity required for one ton of refrigeration will also vary.

The expansion pressure always varies with the temperature to be maintained. The following table of temperatures, expansion pressures and cylinder capacities will be useful in estimating the capacity of compressors:

Lowest Temp. to Be Maintained. Deg. F.	Approx. Temp. of Suction Gas. Deg. F.	Suction Pressure Gage. Lb.	Cu. In. of Cylinder Per Minute.
— 5	—17	5	10,700
0	— 8	10	9,000
+ 10	0	15	7,500
+ 15	+ 6	20	6,000
+ 20	+ 11	25	5,400
+ 25	+ 17	30	4,800

In this table, for convenience, the temperature of the condensing water is assumed to be 70 deg. F. For every 5 deg. variation in the temperature of the con-

densing water above or below 70 deg. F., add or deduct 1 per cent to or from the number of cubic inches of cylinder capacity given.

In every case the lowest temperature to be maintained in any part of the plant must be taken to estimate the required cylinder capacity.

To find the refrigerating capacity of a compressor at any given pressure, multiply the area of the piston, in square inches, by its stroke, in inches, and the number of strokes per minute, and divide by the number of cubic inches given opposite the gage pressure in the table above. The result will be the capacity of the compressor in tons per 24 hr.

612. Q.—What are the consecutive steps in the production of distilled water for the can system of ice-making? What is the purpose of each step?

A.—The usual process of distillation is as follows: The exhaust steam after leaving the engine passes to the oil separator where all oil and any priming water that the steam may contain are separated out. The steam then passes to an exhaust-steam condenser, and any surplus of steam not immediately condensed is taken care of by a back pressure valve, which is usually set at about 2 or 3 lb. above atmospheric pressure. A small vent cock is also provided in the line between the exhaust steam condenser and the back pressure valve to relieve the steam pipe of any air or other undesirable gases. The condensed steam then passes to a reboiler, where it is reboiled by means of live steam to bring any impurities to the surface which are then skimmed off and taken away through a waste pipe. The reboiled condensed steam now passes to a hot filter and then goes to a distilled-water cooler, where the temperature is reduced to that of the cooling water and then enters a fore-cooler, where it is cooled to a temperature of 45 or 50 deg. It passes, next, through a cold water filter and then is ready to be discharged into the cans.

613. Q.—What are the first symptoms when a refrigerating machine is overloaded? We use direct expansion, no brine tank.

A.—Return gases will come back to the machine too hot, because of refrigerating coils having to circulate ammonia too fast; pressures will run up too high; frost

will drop off of refrigerating coils in the chill rooms; temperatures will go up too high, and machine will run too hot.

614. Q.—How much clearance is necessary in ammonia compressors?

A.—The clearance possible is largely a matter of mechanical construction, but generally the effort is made to keep it down to 1/32 in.

615. Q.—What are the relative advantages of single and double acting compressors?

A.—The single acting ammonia compressor is usually of the vertical type and compresses gas on one side of the piston only, that is, on the upper side or the side farthest away from the stuffing box. The gas has at all times access to the space in the cylinder below the piston; here only the suction pressure acts constantly on the under side of the piston. In this form of compressor, the pressure on the stuffing box side is always low, and it is, therefore, easy to keep the piston rod packed. As the work of compression is done on one side of the piston only, the water jacket has a greater opportunity for reducing the temperature of the cylinder walls, and the colder the compressor cylinder is kept, the greater will be the weight of gas pumped per revolution.

The loss due to superheating is therefore less in the single than in the double-acting type. Experiment and test have proved this; the loss due to superheating of single-acting compressors varies from 15 to 22 per cent, while with a double-acting machine it amounts to 25 or 30 per cent.

The double-acting compressor for a given capacity is cheaper to build than the single-acting, as with the same cylinder capacity, it will deliver double the quantity of ammonia. The loss due to superheating, however, is greater, as mentioned above. The adjustment for clearance is a more difficult task with the double-acting compressor, for, if the clearance is cut down on one side, it will be increased by a similar amount on the other side. It is necessary, therefore, to be very careful in the selection of gaskets, to have them of the proper thickness so that the clearance spaces will be reduced to a minimum on both sides of the piston. In the single-acting type, it is merely necessary to adjust the piston so that it just

clears the upper head of the compressor, and this can be accomplished regardless of what the stroke of the compressor may be.

616. Q.—What would be the effect of excessive clearance in the cylinder of an ammonia compressor?

A.—In the mechanical action of a compressor having too much clearance, there would probably be nothing to show any irregularity. The pressure during compression would rise more slowly than it should, and the amount of gas discharged each stroke would be less than it ought to be. Also, the suction pressure would not get down to the point of drawing in fresh gas as quickly as it should, after the beginning of the suction stroke. The result would be a lessening of the capacity of the compressor, but the mechanical action would not be likely to show any particular variation from ordinary.

617. Q.—What would cause an ammonia compressor to burst?

A.—Bursting of an ammonia compressor cylinder is of rare occurrence. It is usually due to some mechanical defect or negligence of the operator. Such an accident might be caused in the following ways: (a) By breaking of suction valve and its falling into the cylinder; (b) breaking of the piston nut; (c) starting up the compressor with discharge valve closed; (d) by flooding the cylinder with liquid. Most accidents of this kind may be traced to mechanical defects, such as crystalizing of valve stems, and from long use or wear—some of which could have been avoided through proper care by the operator.

618. Q.—We have two Frick ammonia compressors, one a 12-ton and the other a 6-ton machine.

Can I run both machines by connecting the suction line from the 6-ton machine to two of my rooms and let the other operate on the rest of the rooms and have both machines discharge into the same discharge line to the condenser and then take ammonia from the one receiving tank to all of the rooms? Or would it do to run both machines on the same suction and discharge lines?

A.—There is no reason why you should not be able to run your ammonia compressors drawing from the

same suction, and discharging to the same pipe, if they are speeded so as to give the same final pressure.

You could run them either according to your first suggestion—taking the suction of one machine from two rooms, and the other from the other rooms, and discharging both into the same condenser; or you could connect both rooms to the same suction, and discharge both machines into the same high-pressure line.

We believe you would get better control of your rooms if you operate the suctions separately, letting them discharge into the same high-pressure line to the condenser, and taking the ammonia for your two rooms from the same receiving tank.

619. Q.—We have two ammonia refrigerating machines which, according to the manufacturers' rating, have capacities of 75 and 25 tons respectively. The former machine is double-acting, has a 15 by 30-in. cylinder and makes 40 r.p.m.; it has a back pressure of 27 lb. and a head pressure of 180 lb. The 25-ton compressor has two 8½ by 23-in. cylinders, is single-acting and makes 50 r.p.m.; back and head pressures are the same as in the case of the first unit.

How may I obtain a check on the capacities of these machines?

A.—The capacity of an ammonia refrigerating machine depends on the number of pounds of ammonia handled per minute and the pressures between which the compressor operates. The higher the back pressure the greater will be the weight taken in for a given size of cylinder, and the lower the head pressure the less will be the heat which must be extracted in the condenser. The calculation of compressor capacity, therefore, resolves itself in two parts; first, determining the refrigerating effect of a pound of ammonia working between the head and back pressures; second, determining the number of pounds of ammonia that the compressor can handle.

For the first, $r = h_1 - (t - t_1)\ s$;

where s is the specific heat for ammonia, taken as one;

t is the temperature at head pressure for saturated ammonia;

t_1 and h_1 are the temperature and heat of vaporization at the back pressure, for saturated ammonia.

These values are found from a table of Properties of Saturated Ammonia, having given the head and back pressures. At 180 lb. pressure, the temperature is 94.3; with a back pressure of 27 lb., the temperature is 14.4, and the heat of vaporization is 560.9. The value of r, therefore, works out to 481 B.t.u. absorbed per pound of ammonia handled.

To get the refrigeration capacity of the compressor, we multiply the cubic feet capacity of piston displacement per minute by the refrigerating value of one pound of ammonia as already calculated, and divide by 200 times the specific volume at back pressure.

The cubic capacity of your 75-ton machine is 245.5 cu. ft. per minute, and the capacity of your 25-ton machine is 75.7 cu. ft. per min. The specific volume at 27 lb. pressure is 6.67. We have, then, the refrigerating capacity for the 75-ton machine figures out to 88.5, and for the 25-ton machine, to 27.1, for the pressures as given in your problem.

Roughly, when corrected for the effect of clearance, a compressor will have about one ton capacity in 24 hr. for each 3 cu. ft. of piston displacement per minute. Figured on this basis, your 75-ton compressor would have a capacity of 81.8 tons, and your 25-ton compressor, a capacity of 25.23 tons. This is only a rough check, however, as it does not take account of the back pressure and head pressure.

620. Q.—In regard to liquid ammonia injection for keeping compressor and stuffing box cool, is it profitable to do this or frost back from the brine tank? Would it be safe to drill and tap a 1-in. tee for ¼-in. pipe and drill and tap the side of the suction valve which is 1½ in. for the same size? I am using metallic packing and must keep the stuffing box cool.

A.—Liquid ammonia injection to keep the compressor cylinder and stuffing box cool is a usual and economical method of accomplishing the result. It is probably better, on the whole, than frosting back from the brine tank, although there is no objection to the latter if the distance from the brine tank to the compressor is not too great.

We would think it unwise to drill and tap a fitting or valve to take out an ammonia line. It is almost impossible to get a threaded joint of this kind which will

hold against ammonia pressure for any length of time. The proper method would be to put in a cross in place of the tee, and take your outlet from the side opening in the cross; then put in a tee one side or the other of the valve, and make the connection to this tee, using regular ammonia fittings at all points.

621. Q.—How is head pressure on an ammonia compressor controlled?

A.—The head pressure on an ammonia compressor may be controlled either by the speed of the engine or by the expansion valve. Speeding up the engine or closing the expansion valve would raise the head pressure, and vice versa.

622. Q.—We have an 8-ton York dry compression refrigerating machine operated by a 20-hp. gas engine running at a speed of 120 r.p.m. This set has given us perfect satisfaction for our cooling purposes; but, due to the fact that the gas supply is inadequate, we are desirous of using an electric motor instead of the engine. What size of machine should we get?

I carry 15 lb. back pressure and the high pressure varies from 175 to 190 lb.

A.—We judge that a 20-hp. motor will be about right for driving your 8-ton compressor. With a back pressure of 15 lb. and a head pressure of 180 lb., you would require about 1.15 hp. per ton for compressing the ammonia.

For an 8-ton compressor, this would require 9.2 hp., and allowing for a 50 per cent over-all efficiency of the compressor, this would mean 13.8 hp. to drive the compressor. With an efficiency of 75 per cent for the motor, this would give 17.25 hp. for the motor, and allowing a 20-hp. motor would give you ample margin to take care of overload.

623. Q.—I have a 25-ton refrigeration plant in which the mysterious disappearance of ammonia cannot be accounted for.

Last winter the plant was overhauled for this trouble by an expert and a complete new charge put into the system. A few months later the receiver could not be made to show any liquid at all. After charging 350 lb. into the system without results, I heated the condensing coils with burning waste and shortly my receiver was

filled; but the next day my charge disappeared again and I had to repeat the operation to get it down and could not keep it in circulation until I cut out two of my five stands of condensing coils. My charge worked well for a couple of months and then gradually disappeared again. Since then, I have charged 500 lb. at intervals into the system with only a temporary effect. I have pumped down the condensers while under a temperature which I produced by pumping boiling water over the coils in order to facilitate the movement of the ammonia and have pumped the coils completely empty and still find no trace of the ammonia.

When I pump down the expansion coils, the back pressure does not gradually rise again, which leads me to suppose that the ammonia is not trapped in the expansion coils.

Could the ammonia be trapped in the expansion coils and not show on the back pressure gage? If so, how could I remove it from there?

My condensing coils are made of galvanized pipe. May this produce an effect on the ammonia? My plant has as few leaks as any I have seen, so this cannot account for the loss of ammonia. What is the trouble?

Would you advise the installation of an ammonia purifier?

A.—The real cause of the trouble is the galvanized coils of the ammonia condenser, provided that the pipe is galvanized on the inside as well as on the outside. Ammonia acts on the zinc and the result is a very fine dust which looks like dry cement. If oil is allowed to enter the system, this dry dust becomes a gum-like mass and will lodge in places and stop the circulation of the ammonia.

Most likely such blocking will be found at the outlet valves of the condenser coils. The best thing to do is to pump out one stand of the condenser at a time and disconnect both inlet and outlet. Make air and steam connections so that the coil can be blown out with both. First use the air. If the coils are galvanized on the inside a cloud of white dust will be blown out. If steam is used first, the condensation will wet this dust and cause it to cling to the pipe and no amount of blowing will loosen it till it becomes dry.

After using the air, next use steam and let the steam blow through till it comes out dry as possible. Then

while the coil is still hot, blow with air again till the outlet end of the coil shows perfectly dry.

If it is true that the condenser coils are galvanized on the inside, this blowing out with air will have to be done about once a month to remove the zinc as it becomes loose. Steam is not necessary unless the coils are coated with oil.

This winter, blow the entire system with air, steam and air. This thorough cleaning may stop the action caused by the combining of ammonia and zinc. If the eating continues, the only remedy is to take out every piece of galvanized pipe and replace with black pipe.

The eating action of ammonia on anything galvanized can be plainly seen in the case of a leaking coil in an ice tank. The galvanizing will come off the ice cans and in time the cans will begin to pit and leak.

If there is no relief gas line between the top of the liquid receiver and the top of the condenser, one should be put in; relieving the light gas from above the ammonia in the receiver will help the circulation greatly.

An ammonia purifier is a great advantage to any refrigerating plant; for, by its use, the impurities can be drawn off and the ammonia kept in a pure and dry state such as is necessary to good results.

Examine the inside surfaces of the condenser pipe for pitting, as the eating action may have begun on the iron. If it has begun, new condensers will be necessary.

624. Q.—I HAVE two freezing tanks containing 10,400 ft. of 1¼-in. pipe and two 4-in. headers.

On account of the carelessness of ice pullers, I have had four break-downs in these tank coils since they were fully charged. This is a flooded system and the loss consisted of the entire capacity of the coils and always occurred on the pipe threads clear of the flanges on the bottom pipe.

About how many pounds of ammonia do you estimate I lost?

A.—It would be a good deal of a guess to say just how much ammonia you lost from your coils by a break which took out all the ammonia in them. In the flooded system, the ammonia may be carried at almost any height in the coils, and the level at which the ammonia happened to be standing would be the deciding factor in determining what the loss was.

For 1¼-in. pipe, it takes 96.25 lineal feet of pipe to contain a cubic foot. Your 10,400 ft. would, therefore, contain 108.2 cu. ft.

You do not give the length of the 4-in. headers, but for 4-in. pipe it takes 11.3 ft. for a cubic foot, and if you had, say, 22 ft. of 4-in. pipe, it would amount to practically 2 cu. ft. of ammonia. This would give 110 cu. ft. total.

At 10 deg. F., ammonia will weigh about 40.2 lb. per cubic foot, so that 110 cu. ft. would give 4422 lb., which would be lost from the 22 ft. of 4-in. headers and 10,400 ft. of 1¼-in. pipe, if the pipe and headers were full, which they probably were not.

In connection with this matter, you should have an inlet and outlet valve on each coil leading from the header, so that in case of a break in the coil, that coil can be shut off from both inlet and outlet headers; or in case of a break at the header, all other coils can be shut off, and the inlet and outlet to the headers shut off, thus losing only the ammonia in the broken coil and the headers.

625. Q.—In absorption refrigeration systems what causes a "boil over" and how may that be prevented?

A.—In regard to trouble with "boil over" in an absorption system, this may be due to any one of several causes. If the rectifier is not getting enough water, it will result in making wet gas, then the tank will act as an absorber, and the generator charge will be weakened so that the liquid will "boil over" faster than the drain pipes can take care of it.

The remedy for this is to cut down the steam on the generator, open the expansion valve to the coils wider, blow out the rich liquor in the tanks and get the ammonia back to the generator and absorber. It is a process which will take some time and patience.

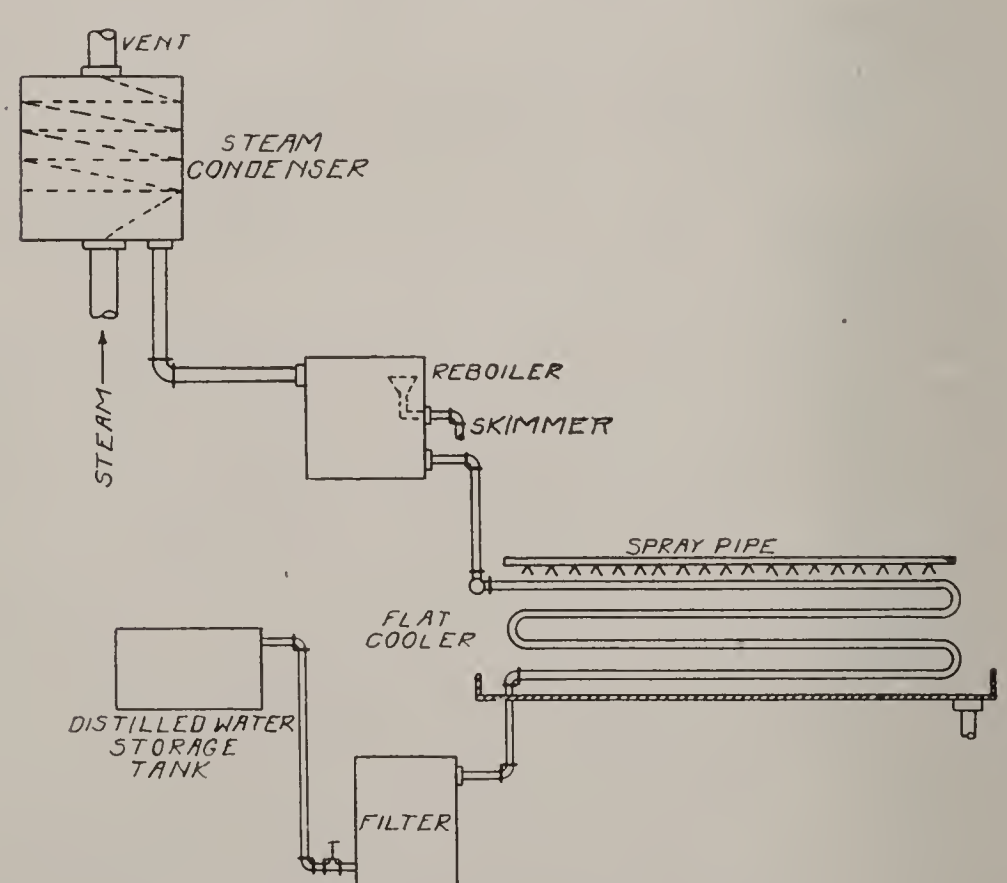
It may be that the analyzer, if you have a live steam type, is not working well, and that this is permitting too much moisture to pass on into the rectifier; or it may be that the separator between the rectifier and the condenser is not taking out the moisture properly.

Another trouble may be that the charge in your system is too weak. This will have to be strengthened, of course, by adding more ammonia. Your rich liquor should test up at about 26 deg. Beaume by the hy-

drometer, and your poor liquor, 17 to 18 deg. If the poor liquor is coming back richer than this, and the rich liquor is showing weak, there is probably a leak in the exchanger, which permits the rich liquor to work back into the poor liquor side. If you find this condition, the remedy, of course, will be to take down the exchanger and get the leaks out of it.

626. Q.—We have an ice machine of 1000 lb. (½ T.) capacity. Water for the ice is condensed from a high-pressure steam jet in a galvanized iron tank. Freezing tank temperature is kept at 10 deg. F., pulling a few cakes of ice at a time, so as to raise the temperature of the tank not over 2 to 4 deg. Ice from cans first put in (while cooling down the freezing tank), comes out clear, but from cans which replace these, it is cloudy. Are we freezing too fast? What is the proper temperature for the freezing tank? How long should be taken to freeze the ice? How can cracking of the cakes when removing from the cans be avoided? Some of the ice had a taste like rotted wood. What was the cause?

A.—Figure A626 is a sketch of a complete distilled water system, and no matter how small your plant may



A626. DISTILLED ICE SYSTEM

be you will find it a hard matter to make good can ice without this system. The water which has gone through the process of distillation, condensation, reboiling, and skimming will not always make clean ice. Water contains a great deal of mineral matter and the white or milky ice is often caused by foul and dirty boilers which prime, throwing over into the condenser undistilled water either as wet steam or water direct. This will cause white ice or large cores in the blocks which have a very offensive odor and a bitter taste sometimes compared to the bitter water from salt works (or as you say, a rotten taste). We should advise you not to take steam from a dirty or crowded boiler, and also to blow off about two gages of water from boiler at least twice in 24 hr. Also you should keep the cooling water used over the steam condenser cut down until the steam blows through the steam condenser at the vent with considerable force. This will purge or carry off a great deal of the objectionable odor and air with it. The reboiler should be equipped with a closed coil, and boiled vigorously.

Low temperature will sometimes cause a white or cloudy film on the outside of ice blocks and will cause ice to break badly. To overcome the breaking of the blocks you should pull the cans and let them stand on the tank for about 30 min. to temper before putting them in the dip tank or sprinkling dump. The usual temperature carried in the freezing tank is 14 deg. F., and to freeze a 200 lb. block of ice at this temperature will require from 48 to 60 hr., depending upon the circulation in the brine tank. A good strong circulation will shorten the time required to freeze out the ice.

627. Q.—Where does the expansion of ammonia take place in the direct expansion system?

A.—Expansion of ammonia takes place through the expansion valve into the cooling coils, whether the system is direct or a brine system. The difference between the brine system and the direct is that in the case of the brine system, the ammonia expansion coils are located in a brine tank and serve to cool a body of brine, which is kept in circulation through the pipes in the refrigerating rooms and ice tanks. In the direct expansion system, the ammonia is taken direct to the cooling rooms and the ice tanks, and the ammonia ex-

pansion coils are placed in these rooms, no brine being used.

628. Q.—I WOULD like to know how to proceed to remove the ammonia from the generator, the condenser, the brine cooler or the absorber of an absorption system. Is there any way in which this can be pumped out and stored in some other part of the system as you would in the case of a compression system?

A.—Getting ammonia out of an absorption system, is not frequently done, but you can clear the system by the following method:

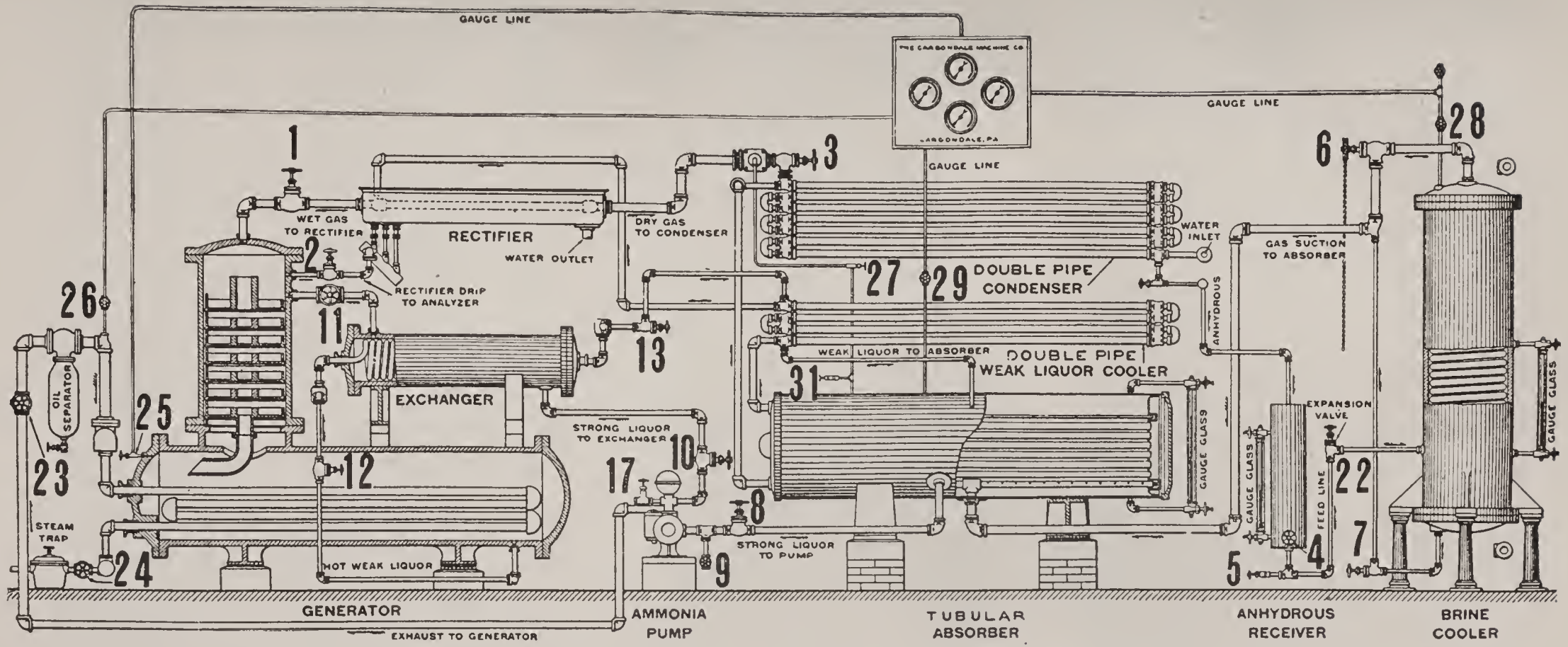
First, close off the rich liquor supply to the exchanger or equalizer, and run steam in the generator for an hour or more to get the liquor as weak as possible. Then close off the gas inlet to the condenser or rectifier, and by putting in cross-over connections so that the pump can draw from the generator and from the exchanger and deliver to the absorber, all the liquor can be pumped to the absorber.

Now, to clear the condenser of gas—first, condense all possible gas by running the condensing water over it; then, after the liquid gas is all in the receiver, and the liquor has all been pumped into the absorber, close off the connection from the condenser to the receiver, open the connection from condenser to generator, and pump out all gas possible, forcing it into the absorber. This will leave the generator, exchanger and condenser sufficiently free from gas so that they may be opened up.

Another method would be to boil out the liquor well in the generator, and then drain the weak liquor into a tank or drums, and pump all the gas into the absorber. Or, if it is desired to work on the absorber as well as the exchanger and generator, the ammonia could be shut back into the receiver and brine cooler, the liquor circulated from absorber to generator and exchanger, and boiled until all the liquor was weak, and the entire liquor system drained to a tank or drums. The ammonia gas would then be pumped out with a vacuum pump and forced into a drum.

629. Q.—COULD I obtain better results from a shell brine cooler in an ice tank by using a needle point expansion valve instead of a 1-in. flat disc angle valve?

A.—No better results will be obtained by the use of a needle point valve. In the first place, the needle point



A628. DIAGRAM OF ABSORPTION SYSTEM

valve is not made as large as 1 in. This style of valve is used for feeding ammonia to coils where close regulation is required. In short coils or in coils in extremely cold rooms, very close regulation is necessary or the coils will be flooded and cause liquid to be drawn into the compressor.

The shell type brine cooler as installed in ice tanks is built on the so-called flooded system and the amount of ammonia fed into it can easily be properly handled with a regular globe or angle valve. The accumulator will take care of any excess feed within a reasonable amount.

It is best to leave the cooler connected up as it is with the 1-in. angle valve, and by a little practice and by noting results, the proper amount of opening will soon be learned.

630. Q.—What is a practical method of defrosting a refrigerating coil?

A.—If direct expansion coils are used, pump all ammonia liquid out of the coils; then pump a little hot gas from the machine very slowly through the coils. If brine coils are used, get all brine out of the coils; then turn a little steam in, or if you have time and can spare the room or coils, turn some jets of steam on the outside of the coils; if you have plenty of exhaust steam to spare, that will do and will be cheap.

Chloride of calcium will defrost coils if it is sprayed over the coils, and can be reused, but it is rather expensive.

631. Q.—How may the required temperature of cooling water for use in connection with an ammonia condenser be determined?

A.—Knowing the head pressure carried, it is necessary to secure the corresponding value of temperature of the ammonia which may be obtained from a table of the properties of saturated ammonia. It is usual, then, to carry the temperature of the cooling water about 10 deg. lower than this.

632. Q.—Is the double-pipe ammonia condenser more efficient than the atmospheric type?

A.—The double-pipe condenser is usually considered more efficient than the atmospheric type, because of the better contact secured of the cooling water against the ammonia piping. The only advantage of the atmospheric

type is the ability of the condenser to evaporate some of the cooling water, the evaporated moisture being carried away by the air currents. The double-pipe condenser is, of course, more expensive than the atmospheric, but so far as efficiency is concerned, we do not believe that there is very much difference, or that the water consumption would be materially affected.

633. Q.—How would you thaw out a frozen brine cooler?

A.—The general method employed in thawing out a frozen brine cooler is to have the system so connected that either hot or cold ammonia may be circulated through the piping. By the use of hot ammonia the cooler can be thawed out.

634. Q.—If the average difference in temperature between the intake brine pipe and the outlet pipe in a storage room is 5 deg. and the amount of brine pumped is 15 gal. per min., how would you find the amount of heat taken up by the brine?

A.—The specific heat of brine depends on the per cent salt in the mixture. For a 5 per cent solution, which would be used where the cooling is above 30 deg., the salinometer would show 20, and the specific heat is 0.96—that is, each pound would take up 0.96 B.t.u. per degree temperature difference. For a 10 per cent solution, to be used where the temperature is to be held at 25, the salinometer would show 40, and the specific heat is 0.892. For a 20 per cent solution, where the temperature is to be held at 10 deg., the salinometer would show 80, and the specific heat is 0.829.

To find the amount of heat taken up, you would multiply your temperature difference by the number of gallons pumped per minute, by the weight of a gallon, and by the specific heat. For instance, if you are using a 10 per cent solution, the weight per gallon is 8.97 lb., so that your problem would be $5 \times 15 \times 8.97 \times 0.892$, which works out to about 600 B.t.u. per minute.

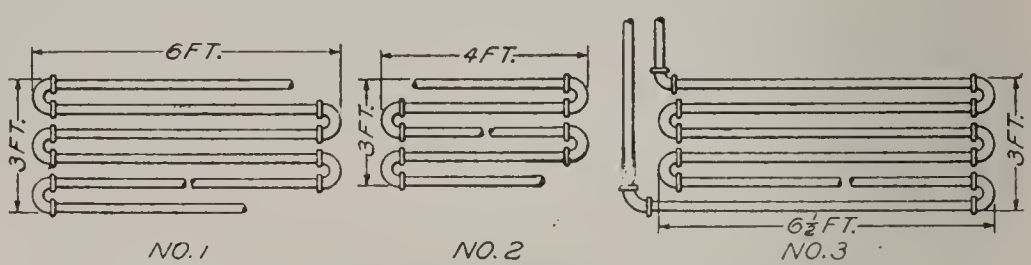
635. Q.—We have a 2-ton ice tank located about 70 ft. from a small cold storage room cooled by brine pumped from this tank. The connecting pipes are underground and are uninsulated. In wet or damp weather, we are unable to keep the room cool, and a

continuous stream of warm brine runs back into the ice tank.

The coils, shown in Fig. A635, are made up of 1-in. pipe. Coil No. 1 is placed on the left side of the room, No. 2 on the back end, while No. 3 is placed in a bunker on the right side.

Will I gain anything by cooling by direct expansion with the expansion valve in the storage room?

Can I use a ¼-in. pipe to carry the liquid over and a ¾-in. pipe as a return?



A635. ARRANGEMENT OF BRINE COILS

A.—The installing of a direct expansion system for cooling the cold storage room would necessitate the continuous operation of the ice machine, or the temperature would vary greatly. In a small plant, even when a small amount of ice is made, the compressor is usually shut down during at least part of the night, and the brine pump kept going for holding the temperature of the rooms and ice vault. If the compressor is run all the time, there is always the liability of a breakdown, and when this occurs, and the cold storage is piped for direct expansion, the refrigeration stops when the machine stops. With brine circulation, the cold storage can be held at a safe temperature for several hours after the machine is stopped.

If direct expansion is desired, it is best to use new pipe instead of the old brine coils, as they will be hard to clean of scale, dirt and moisture. Make one coil covering the 3 walls. The 3 small coils now in use are so short that it would be found difficult to regulate the ammonia, and the brine coils would have to be all taken down and the joints made up with litharge and glycerin. New pipe and fittings will be found better. Use a ½-in. liquid line and 1-in. suction; run them close together, and cover them in one covering of either cork or hair felt. The liquid can be admitted either at the top or bottom of the coil, as the results are about the same with proper handling.

For a small cold storage room, the brine circulating system is better. As we do not know the size of the room, or the temperature at which it is held, or the amount of produce handled, we cannot tell whether the amount of coil surface is sufficient.

The brine lines from the pump to the room and back to the ice tank should be taken out of the ground, if possible, and run overhead or along the wall. They should be covered with the best insulating material available. The insulation will pay for itself in a season. If it is not possible to run these lines except underground, they should be covered with moisture-proof insulation and a box built around them. Hemlock planks make a good box which will last for years.

636. Q.—What refrigeration capacity is required to cool 1000 gal. of milk an hour from 60 deg. F. to 38 deg.? The milk runs over copper coils in which brine is circulated.

A.—As milk has a specific gravity nearly that of water, we will consider the specific heat as one; and 1000 gal. equals 8333 lb. an hour to be reduced from 60 deg. to 38 deg., or a temperature drop of 22 deg. Multiplying the specific heat, 1, by 8333 (the total weight of the liquid), and again by 22, gives us 183,326 B.t.u. to be extracted from the liquid per hour. One ton of refrigeration will extract 12,000 B.t.u. per hour, therefore 183,326 divided by 12,000 gives us 15.28 tons, as the required refrigerating capacity.

637. Q.—What amount of refrigeration is needed to cool an ice storage house down to 24 or 26 deg. F.?

A.—The ice itself will require but little refrigeration to hold it at 24 deg. so that the main consideration is to carry off the heat coming in through floor, walls and roof. Heat leakage is rated as the B.t.u. per 24 hr. per square foot of surface per degree difference between inside and outside temperatures. For ordinary construction it is safe to allow 4 B.t.u.

First, find the square feet of surface in floor, walls and ceiling; multiply this by the temperature difference between the room and outside, and by 4. The product will be the B.t.u. to be carried off in 24 hr. Divide by 288,000, the B.t.u. for 1 T. refrigeration, and the quotient will be the tons capacity required for leakage.

For the ice, you must remove, say 40 B.t.u. per cubic foot per 24 hr. to hold it at the temperature desired, with the ice in dry condition. Multiply the cubical contents of the room by 40 and divide by 288,000 to get the tons refrigerating capacity to hold the ice cold when the room is full.

The sum of the two capacities will be the plant capacity required to cool the storage room.

638. Q.—What is the temperature best suited for the family refrigerator? Also the highest and the lowest temperature permissible for the preservation of food?

A.—The temperature that is considered best on the average is 34 deg. F., although comparatively few refrigerators of the ordinary kind hold a temperature as low as this. Usually the temperature is around 40.

So far as the high and low temperatures for this purpose are concerned, herewith is a table of the high and low temperatures for different materials, which comes as near answering the question as anything available.

	High Deg. F.	Low Deg. F.
Fruit	40	34
Vegetables	38	32
Fish	38	30
Dairy Products	36	32
Meats	40	30

PART VII

Miscellaneous

Compressed Air

639. Q.—An Ingersoll-Rand high-pressure electrically-driven air compressor has dimensions as follows: High-pressure cylinder, 20¼ by 24 in.; low-pressure cylinder, 32 by 24 in. The speed is 167 r.p.m. How many cubic feet of air can this compressor deliver per minute?

A.—The formula for finding the capacity of an air compressor is as follows:

$$0.7854\ d^2\ L\ N \times VE \div 1728 =$$

capacity in cubic feet of free air per minute. In which d is the diameter of the low-pressure cylinder in inches. L is the length of the stroke. N the number of strokes per minute and VE the volumetric efficiency. Working this out from your problem we find that it is 3705 times the volumetric efficiency. The volumetric efficiency depends largely upon the clearance space at the end of the cylinder, the speed of the compressor and the type of valves employed. It will range all the way from 80 to 95 per cent. Assuming 90 per cent volumetric efficiency, your compressor would have a capacity of 3345 cu. ft. of free air per minute, approximately.

640. Q.—Is steam cylinder oil safe to use in an air compressor cylinder?

A.—Cylinder oil as commonly used for lubricating steam cylinders is not safe with air compressors, because the oil is apt to vaporize, and sometimes causes explosions in the receiver. An effective lubricant for air compressors is soapy water, which is recommended by some manufacturers of air compressors.

641. Q.—On an indicator card from an air compressor, what indicates leaky valves?

A.—Detection of leaky air compressor valves from an indicator card is done simply by knowing the shape of compression curve that you ought to get, which is nearly the same as the expansion curve on a steam engine. If the admission valves are leaking, the air will be forced back out of the admission, and the compres-

sion curve will rise more slowly than it should. Also, the expansion curve which precedes the opening of the admission valves will drop more rapidly than proper. It is likely, also, that if the discharge valves are leaking, the admission pressure will be higher than normal, because some air will be leaking back from the high pressure lines into the cylinder during admission, and the pressure will rise in the compressor faster than it ought because of the leakage back into the cylinder of high pressure air from the discharge lines.

642. Q.—In a 2-stage compressor delivering 1000 cu. ft. of air per minute, at 100 lb. gage, how many cubic feet of free air would be taken in, intercooler pressure being 28 lb. gage?

A.—Due to the fact that in an air compressor you cannot consider the compression as purely adiabatic, neither as purely isothermal, we will, in the case of a 2-stage compressor, attempt to strike an average value by considering the compression in one cylinder adiabatic and in the other isothermal.

In your question, you state that the final volume of the air is 1000 cu. ft., at a pressure of 100 lb. gage, and also that the intercooler pressure is equal to 28 lb. gage. We, therefore, have for the second cylinder, a final volume equal to

$$V_2 = 1000 \text{ cu. ft.}$$

Final pressure, $P_2 = 100 + 14.7$ or 114.7 lb. absolute. The intercooler pressure, $P_1 = 28 + 14.7$, or 42.7 lb. absolute.

We know that when a gas expands, or is compressed isothermally, that $V_2 P_2 = V_1 P_1$ or in the present case,

$$1000 \times 114.7 = 42.7 V_1$$

which by transposition will give us a value for V_1 of

$$1000 \times 114.7 \div 42.7 = 2686 \text{ cu. ft.}$$

This is the number of cubic feet which will have to be delivered to the second cylinder under a pressure of 28 lb. gage in order to deliver 1000 cu. ft., at a pressure of 100 lb. gage.

We now have, in taking up the first cylinder, a value of 2686 for V_2, 42.7 for P_2, 14.7 for P_1, and will solve for V_1. When a gas expands or is compressed adiabatically, we have

$$P_1 V_1^{1.41} = P_2 V_2^{1.41}$$

which by transposition results in

$$\log V_1 = \log V_2 + \frac{1}{1.41} \log \left(\frac{P_2}{P_1} \right)$$

which in the present case would be

$$\log V_1 = \log 2686 + 0.708 \log 2.9$$
$$= 3.429106 + 0.708 \times 0.462398$$
$$= 3.746484$$

Consulting a table of logarithms, we find that 3.746484 is the log of approximately 5578, or the number of cubic feet of free air required for a delivery of 1000 cu. ft. of compressed air, under a pressure of 100 lb.

643. Q.—COMPRESSED AIR at 100 lb. gage flows from a hole ¼ in. diam. to atmosphere from a pipe line large enough so that there is no appreciable friction loss. How much power will be required per hour?

A.—From a table of air flow, the amount of air escaping from a ¼-in. round hole, at 120 lb. pressure absolute, would be 103 cu. ft. per minute of free air.

From a table of horsepowers, to compress this air, it is found that this requires 0.208 hp. per cubic foot of free air, allowing 15 per cent for friction. This would mean that $103 \times 0.208 = 21.42$ hp. would be required.

If the flow continued for an hour, you would have used 21.42 hp.-hr., or 42,372,000 ft. lb. of energy.

644. Q.—A VERTICAL cylinder containing air at a temperature of 40 deg. F. is fitted with a piston 6 in. in diameter. If a pressure of 500 lb. on the piston is required to keep it 18 in. from the bottom of the cylinder, how much pressure must be added so that it will be in the same position when the temperature of the air is 85 deg. F.?

A.—In solving this problem, we will make use of a formula based upon a corollary of Gay-Lussac's law that if the volume remains constant, increase of temperature produces rise in pressure of gas proportioned to the increase of absolute temperature. The formula is:

$$p_2 = p_1 \left(\frac{460 + t_2}{460 + t_1} \right)$$

where p_1 = initial pressure
p_2 = final pressure
t_1 = initial temperature, F.
t_2 = final temperature, F.

In this example, $p_2 = 500 \left(\frac{460 + 85}{460 + 40} \right)$

$p_2 = 545$ lb.

Hydraulics

645. Q.—Explain by rule and example how to find the amount of water that will flow through a given pipe under varying pressures.

A.—The formula for determining the quantity of water that will flow through a given pipe under various pressure heads is

$$Q = a\,c\,\sqrt{r\,s} \qquad \text{(Kent)}$$

where Q = quantity of water, in cubic feet per second;
a = area, in square feet, of pipe cross-section;
c = coefficient determined from experiment;
r = mean hydraulic radius (area of wet cross-section divided by wet perimeter);
s = slope (pressure head divided by length of pipe).

Example:

Find the quantity of water which will flow through 100 ft. of 2-in. wrought-iron pipe under a pressure head of 40 ft. (equivalent pressure $= 40 \times 0.433$ or 17.32 lb. per square inch).

The internal area of a 2-in. pipe is 3.355 sq. in. or 0.02315 sq. ft. For clean 2-in. pipe, c may be taken as 93. Kent gives the value of $\sqrt{r}$ as 0.204 for 2-in. pipe running full; s will be $40 \div 100 = 0.4$, and $\sqrt{s} = 0.631$. Then we have,

$$Q = 0.0231 \times 93 \times 0.204 \times 0.631$$
$$= 0.2765 \text{ cu. ft. per sec.}$$

646. Q.—A certain pump delivers 100 gal. of water a minute. What would be the velocity of flow through a 3-in. delivery pipe? The actual internal diameter of the 3-in. pipe is 3.07 in.

A.—A pump delivers 100 gal. of water per minute, or $100 \times 231 = 23{,}100$ cu. in. (231 cu. in. equal 1 gal.). Actual diameter of 3-in. pipe equals 3.07 in. Then area of pipe equals $3.07^2 \times 0.7854 = 7.4$ sq. in.

Then $23{,}100 \div 7.4 = 3121$ in. velocity per min.

$3121 \div 12 = 260$ ft. nearly = velocity flow in feet per minute.

647. Q.—What is the rule for velocity and volume of discharge of water under 70 lb. pressure through a 1½-in. pipe 125 ft. long?

A.—There are so many varying conditions affecting the flow of water that all hydraulic formulas are only approximations of the correct result.

The following formulas, however, are applicable to the flow of water in clean iron pipes up to 5 in. in diameter, and will give approximate results:

$$v = c\sqrt{\frac{d}{4} \times \frac{h}{l}}$$

$$q = ac\sqrt{\frac{d}{4} \times \frac{h}{l}}$$

Where v = the velocity in feet per second.
d = inside diameter of the pipe in feet.
h = the head of water in feet.
l = the length of pipe.
q = the discharge in cu. ft. per minute.
a = the internal area of the pipe in sq. ft.
c = a constant depending on the size of pipe.

A 1½-in. pipe has an internal diameter of 1.6 inches, or .134 ft., and an area of .0141 sq. ft. A pressure of 70 lbs. per sq. ft. corresponds to a head of 161 ft. According to D'Arcy, for a 1½-in. pipe, c = 88.1.

Substituting these values in the second formula above, we get:

$$q = .0141 \times 88.1\sqrt{\frac{.134}{4} \times \frac{161}{125}}$$

which, reduced, gives .258 cu. ft. per second. This corresponds to a flow of 15½ cu. ft., or 116 gallons per minute.

The following values of c from Kent are given for pipes up to 5 in. in diameter.

Diam. in.	⅜	½	¾	1	1¼	1½	1¾	2	2½	3	4	5
c	59.4	65.7	74.5	80.4	84.8	88.1	90.7	92.9	96.1	98.5	101.7	103.8

648. Q.—What is the formula for figuring the flow of water through a V-notch weir.

A.—The formula for a sharp-edged notch is:

$$Q. = C\ (4/15\ l\ h\ \sqrt{2\ g\ h})$$

where l is the width of the notch at distance h above

the vertex, and C is a coefficient with a value of from 0.57 to 0.60, based on experiments with heads up to 3 ft. and vertex angles of 28, 60, 90, and 120 deg. The coefficient 0.57 was found for the 60 and 90 deg. weirs, and 0.60 for the 28 and 120 deg. weirs.

In the above formula, Q is the number of cu. ft. per sec., and g is the value of gravity.

649. Q.—What is a simple formula for measuring the amount of water by the use of a weir? The discharge is from a 6-in. pipe into a ditch. We placed a box with an 8-in. wide weir at one end of the box under the discharge and the water flowed over the weir 1½ in. high.

A.—You may readily solve your problem by the application of the Francis formula, which is as follows:

$$Q. = 3.33\ (l - 0.2\ h)\ h^{3/2}$$

where Q is the number of cubic feet flow per second, l is the length in feet, and h is the height, also in feet.

For your problem,

$$Q. = 3.33 \left(\frac{8}{12} - 0.2 \times \frac{1.5}{12} \right) \left(\frac{1.5}{12} \right)^{3/2}$$

which, solved, gives the value of Q as 0.094 cu. ft. per sec.

Belting and Gearing

650. Q.—I wish to operate an electric generator by means of a counter shaft, the driving pulley on which has a diameter of 26 in.; the generator carries a 4-in. pulley and is supposed to operate at a speed of 1900 r.p.m. The pulley on the main shaft which will drive that on the counter shaft is 32 in. in diameter and has a speed of 84 r.p.m. To what size of pulley should this line shaft pulley be belted in order to run the generator at its rated speed?

What size belt should I use on the generator pulley?

A.—With 4-in. pulley on the generator and 26-in. pulley on the counter shaft, the speed ratio will be about 6.5. Your total speed ratio between the generator and the line shaft is 1900 divided by 84, or 22.6. Dividing this by 6.5 gives 3.48 as the speed ratio between the counter shaft and the line shaft. With 32-in. driving pulley, the driven pulley on the counter shaft would be 32 divided by 3.48, or 9.2 in. diameter.

It will be necessary to allow for a certain amount of slip, and if we allow for about 5 per cent slip, this would make your small pulley on the counter shaft 8.5 in. It would be better to have this allowance so that you would have the generator running at a little above, rather than under, speed, as you can cut down the voltage by a field rheostat, whereas if you are under speed, there will be no way of bringing up the voltage.

We cannot tell you what width of belt to use, as you do not say what the capacity of your generator is to be; but that will be taken care of, practically, by the width of the pulley on the generator.

651. Q.—How is the width of belt to transmit a given horsepower found?

A.—The horsepower which a given belt may safely transmit may be obtained by the use of the following formula:

$$\text{Hp.} = 0.00024\ D\ N\ W$$

Where D is the diameter of the pulley, N the number of revolutions per minute of the pulley, and W is the width of the belt, in inches.

652. Q.—I have a motor operating at a speed of 1000 r.p.m. and equipped with a 6-in. pulley which I desire to use in connection with a machine having a 10-in. pulley and which is to be run at a speed of 150 r.p.m. I find, however, that it will be necessary to employ a counter shaft. How may I calculate the speed and pulley sizes required?

A.—In the problem of calculating pulley diameters and speeds, there is one rule which is very important and should be kept in mind at all times. This is, that the diameter of the driven pulley, in inches, times its speed, in revolutions per minute, is equal to the diameter of the driving pulley, in inches, times its speed, in revolutions per minute.

In the problem you present, we have the diameters of the motor and the driven machine pulleys, also their respective speeds, and the problem is to determine not only the speed but also the diameters of the pulleys on the counter shaft. We have, arbitrarily, selected a speed of 300 r.p.m. as that most suitable for the counter-shaft. Then applying the rule given above, we find that the product of the diameter of the motor pulley times its speed is equal to 6 times 1000, or 6000.

The speed of the counter-shaft is supposed to be 300 r.p.m., so that the diameter of the driven pulley on the counter-shaft must be equal to 6000 divided by 300, or 20 in. The speed of the driven machine is supposed to be 150 r.p.m. and the diameter of its pulley is 10 in. The product of these two factors is 10 times 150, or 1500. Dividing 1500 by the speed of the counter-shaft, or 300, we have a driving pulley diameter for the counter-shaft of 5 in. diameter.

653. Q.—What is the best way to glue an 8-in. and a 20-in. two-ply leather belt?

A.—A satisfactory glue for cementing leather belts consists of two parts Page's liquid fish glue and one part Russian liquid isinglass.

Industrial Engineering gives a belt cement recipe calling for equal parts of glue and American isinglass, which are allowed to soak in water for 10 hr. The mixture is then brought to the boiling point and pure tannin added until it has the appearance of the white of an egg. This glue is applied hot to the surfaces of the splice.

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